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Vol. 239 MAY, 1945 No. 5

THEORY OF BENDING, TORSION AND BUCKLING OF THIN-WALLED MEMBERS OF OPEN CROSS SECTION.

BY

STEPHEN P. TIMOSHENKO,

Professor of Theoretical and Applied Mechanics, Emeritus, Stanford University.

PART III.

8. **Buckling by Flexure and Torsion under Central Thrust.**—Let us consider now the general case of buckling where, under central thrust, not only torsion but also bending of the axis of the compressed bar occurs. We assume that x and y are the principal centroidal axes of the cross section of the bar and x_o, y_o are the coördinates of the shear center. The deflections of the shear center axis in the x- and y-directions, we denote by u and v, respectively, and denote, as before, by φ the angles of rotation of cross sections with respect to the shear center axis. Then the deflections of the centroidal axis during buckling, as can be seen from Fig. 26, are

$$u + y_o\varphi \qquad \text{and} \qquad v - x_o\varphi.$$

Assuming that only a thrust P is acting at the ends as in the case of simply supported bars, we find that bending moments with respect to principal axes at any cross section are

$$M_x = P(v - x_o\varphi), \qquad M_y = -P(u + y_o\varphi).$$

The differential equations (17) for the deflection curve of the shear-center axis become

$$EI_y \frac{d^2u}{dz^2} = -P(u + y_o\varphi),$$
$$EI_x \frac{d^2v}{dz^2} = -P(v - x_o\varphi). \tag{56}$$

(Note—The Franklin Institute is not responsible for the statements and opinions advanced by contributors in the JOURNAL.)

To write the equation for the angle of twist φ, we proceed as in the previous article. We take a strip of cross section tds and defined by coördinates x and y in the plane of the cross section. The components of its deflection in the x and y directions during buckling are

$$u + (y_o - y)\varphi \qquad \text{and} \qquad v - (x_o - x)\varphi, \text{ respectively.}$$

Taking the second derivatives of these expressions with respect to z and considering an element dz of the fiber, we find, as before, that the compressive forces σtds acting on the slightly rotated ends of the element dz give the following forces in the x and y directions:

$$-\sigma tds\frac{d^2}{dz^2}[u + (y_o - y)\varphi]dz \qquad \text{and} \qquad -\sigma tds\frac{d^2}{dz^2}[v - (x_o - x)\varphi]dz.$$

Denoting, as before, by primes, the derivatives with respect to z and making the moments of the above forces with respect to the shear-

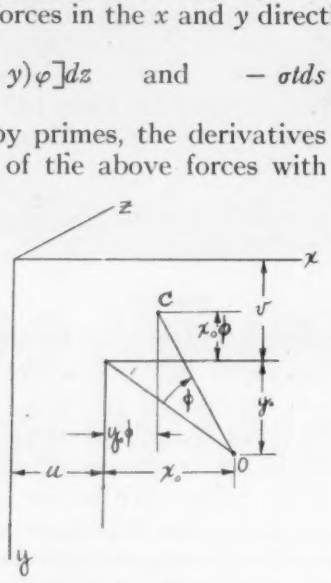

FIG. 26.

center axis, we obtain, as the contribution of one strip, the following torque per unit length of the bar:

$$dm_z = -\sigma tds[u'' + (y_o - y)\varphi'](y_o - y) + \sigma tds[v'' - (x_o - x)\varphi''](x_o - x).$$

Making integration over the entire cross section and observing that

$$\sigma\int_A tds = P, \qquad \int_A ytds = O, \qquad \int_A xtds = O,$$

$$\int_A y^2tds = I_x, \qquad \int_A x^2tds = I_y,$$

we obtain

$$m_z = (x_ov'' - y_ou'')P - r_o^2P\varphi'',$$

where

$$r_o^2 = x_o^2 + y_o^2 + \frac{I_x + I_y}{A}$$

is the square of the radius of giration of the cross section with respect to its shear center. Substituting the calculated value of m_z into eq. (51) we obtain

$$C_1\varphi^{IV} - (C - r_o^2P)\varphi'' - P(x_ov'' - y_ou'') = 0. \qquad (57)$$

This equation together with eqs. (56) will now be used to determine the critical value of thrust P.*

It is seen that the angle of rotation φ enters in all three equations, which indicates that, in the general case, torsional buckling is accompanied by bending of the axis, and we have a combined torsional and flexural buckling. In the particular case when $x_o = y_o = o$, i.e. when the shear-center axis coincides with the centroidal axis, each of eqs. (56) and (57) contains only one unknown and can be treated separately. Equations (56) give then two values of the critical load, corresponding to Euler's buckling in two principal planes. Equation (57) contains, in such case, only one unknown φ and gives us the critical load for purely torsional buckling already discussed in the preceding article. From the three values of the critical load naturally the smallest will be taken into consideration in practical applications.

Returning to the general case, let us assume that the ends of the compressed bar cannot rotate with respect to the z-axis, Fig. 23, but are free to warp and to rotate freely with respect to the x- and y-axes. In such case, the end conditions are

$$\left.\begin{aligned} u = v = \varphi = o, \\ u'' = v'' = \varphi'' = o, \end{aligned}\right\} \qquad \text{for} \quad z = o \quad \text{and} \quad z = l.$$

All these conditions will be satisfied by taking the solutions of eqs. (56) and (57) in the form

$$u = A_1 \sin\frac{\pi z}{l}, \qquad v = A_2 \sin\frac{\pi z}{l}, \qquad \varphi = A_3 \sin\frac{\pi z}{l}. \qquad (58)$$

Substituting these expressions into eqs. (56) and (57), we obtain the following equations for calculation constants A_1, A_2 and A_3:

$$\left.\begin{aligned} \left(P - EI_y\frac{\pi^2}{l^2}\right)A_1 + Py_oA_3 = 0, \\ \left(P - EI_x\frac{\pi^2}{l^2}\right)A_2 - Px_oA_3 = 0, \\ Py_oA_1 - Px_oA_2 + \left(-C_1\frac{\pi^2}{l^2} - C + r_o^2P\right)A_3 = 0. \end{aligned}\right\} \qquad (59)$$

These equations are satisfied by taking $A_1 = A_2 = A_3 = 0$, which cor-

* The system of equations equivalent to eqs. (56) and (57) was first obtained by Robert Kappus; see "Jahrbuch der deutschen Luftfahrtforschung," 1937 and "Luftfahrtforschung," vol. 14, p. 444, 1938.

responds to the straight form of equilibrium of the compressed bar. To have a buckled form of equilibrium eqs. (59) must yield for A_1, A_2, A_3 solutions different from zero which is possible only if the determinant of eqs. (59) vanishes. To simplify the writing we introduce here the notations:

$$P_1 = \frac{\pi^2 EI_y}{l^2}, \qquad P_2 = \frac{\pi^2 EI_x}{l^2}, \qquad P_3 = \frac{1}{r_o^2}\left(C + C_1 \frac{\pi^2}{l^2}\right), \tag{60}$$

in which P_1 and P_2 are the Euler loads for buckling in two principal planes and P_3 denotes the critical load for purely torsional buckling, as given by eq. (54). Then, equating to zero the determinant of eqs. (59), we obtain

$$\begin{vmatrix} P - P_1, & 0, & y_o P \\ 0, & P - P_2, & -x_o P \\ y_o P, & -x_o P, & r_o^2(P - P_3) \end{vmatrix} = 0$$

which, after evaluation, gives the following cubic equation for calculating critical values of P:

$$(-r_o^2 + x_o^2 + y_o^2)P^3 + [(P_1 + P_2 + P_3)r_o^2 - x_o^2 P_1 - y_o^2 P_2]P^2 - r_o^2(P_1P_2 + P_2P_3 + P_1P_3)P + P_1P_2P_3r_o^2 = 0. \tag{61}$$

From this equation one important conclusion can be obtained. Assume that $P_1 < P_2$, i.e. that the smaller Euler load corresponds to bending in the xz-plane, and investigate the sign of the left-hand side of eq. (61) for various values of P. If P is very small, we can neglect all terms containing P and the left-hand side of the expression reduces to $P_1P_2P_3r_o^2$, which is positive. Assume now that P takes the value P_1; then the left-hand side of eq. (61) reduces to $y_o^2P_1^2(P_1 - P_2)$ and, since $P_1 < P_2$, it is negative. This indicates that eq. (61) has a root smaller than P_1 and that taking into consideration the possibility of torsion during buckling we will always obtain a critical load smaller than the Euler load.

To find the critical load in a particular case, we calculate, by using notations (60), the numerical values of the coefficients of eq. (61) and then solve this cubic equation.* We will get three values for P from which the smallest will be taken in practical application. Substituting these values into eqs. (59), we find for each possible critical form the ratios $A_1:A_3$ and $A_2:A_3$. These ratios, establishing the relation between rotation and translation of cross sections, define for each critical form the position of the axis, parallel to the axis of the bar, with respect to which the cross-sections rotate during buckling.

If the bar has very thin wall and short length l, P_3 may become small in comparison with P_1 and P_2. In such case the smallest root of eq.

* This solution can be greatly simplified by the use of nomogram as shown in the previously mentioned paper by Kappus.

(61) approaches the value P_3. Substituting this in eqs. (59), we find that A_1 and A_2 are small in comparison with rotational displacements, which indicates that the buckled form approaches purely torsional buckling discussed in the preceding article. In the case of a thick wall and large length l, P_3 is usually large in comparison with P_1 and P_2 and the smallest root of eq. (61) approaches the value P_1. The effect of torsion on the critical load is small in such case and the usual column formula gives satisfactory results.

If the cross section has an axis of symmetry, the calculation of P_{cr} is simplified. Let the x-axis be the axis of symmetry, then $y_o = o$ and the term containing φ in the first of eqs. (56) vanishes. Buckling of the bar in the plane of symmetry is independent of torsion and the corresponding critical load is given by Euler's formula. We have to consider only buckling perpendicular to the plane of symmetry and torsion. The corresponding equations are

$$EI_x v'' = -P(v - x_o\varphi),$$

$$C_1\varphi^{IV} - (C - r_o^2 P)\varphi'' = Px_o v''.$$

Proceeding as before and using solution (58), we obtain for calculating the critical load

$$\begin{vmatrix} P - P_2, & -x_o P \\ -x_o P, & r_o^2(P - P_3) \end{vmatrix} = 0$$

which gives

$$r_o^2(P - P_2)(P - P_3) - x_o^2 P^2 = 0. \qquad (62)$$

This quadratic equation gives two solutions which, together with the Euler load for buckling in the plane of symmetry, represent the three critical values for P from which the smallest must be used in practical applications. Considering the left-hand side of eq. (62), we see that for very small values of P it reduces to the value $r_o^2 P_2 P_3$ which is positive. We see also that it is negative for $P = P_2$ and for $P = P_3$, since its value then reduces to $-x_o^2 P^2$. It is negative also for all values of P between $P = P_2$ and $P = P_3$, since the first term in it becomes negative and the second is always negative. From this discussion we conclude that one root of eq. (62) is smaller than either P_2 or P_3 while the other is larger than either. The smaller of these roots or the Euler load for buckling in the plane of symmetry gives us the required critical load.

All above made conclusions are based on solution (58). Without any complication we can take the solution in a more general form and assume

$$u = A_1 \sin\frac{n\pi z}{l}, \qquad v = A_2 \sin\frac{n\pi z}{l}, \qquad = A_3 \sin\frac{n\pi z}{l}, \qquad (63)$$

which corresponds to the assumption that during buckling the bar is subdividing in n half waves. Our previous conclusions will hold also

in this case; we have only to substitute into expression (60) the values $n^2\pi^2/l^2$ instead of π^2/l^2. The corresponding critical values of the load will naturally be larger than those obtained for buckling in one half wave and are of practical interest only if the bar has intermediate equidistant lateral supports.

If the ends of the bar are built-in, the end conditions are

$$\left.\begin{aligned} u = v = \varphi = o, \\ u' = v' = \varphi' = o, \end{aligned}\right\} \quad \text{for} \quad z = o \quad \text{and} \quad z = l.$$

Since, during buckling, end moments will appear, we will have, instead of eqs. (56), the following equations:

$$\begin{aligned} EI_y \frac{d^2u}{dz^2} &= -P(u + y_o\varphi) + EI_y \left(\frac{d^2u}{dz^2}\right)_{z=o}, \\ EI_x \frac{d^2v}{dz^2} &= -P(v - x_o\varphi) + EI_x \left(\frac{d^2v}{dz^2}\right)_{z=o}. \end{aligned} \tag{64}$$

These equations, together with eq. (57),* will define now the buckling forms of the bar and the corresponding critical loads. All these equations and the end conditions will be satisfied by taking

$$\begin{aligned} u = A_1\left(1 - \cos\frac{2\pi z}{l}\right), \quad v = A_2\left(1 - \cos\frac{2\pi z}{l}\right), \\ \varphi = A_3\left(1 - \cos\frac{2\pi z}{l}\right). \end{aligned} \tag{65}$$

Substituting these expressions into eqs. (57) and (64), we obtain, for calculating critical loads, the same eq. (61); it is only necessary to use $4\pi^2/l^2$ instead of π^2/l^2 in notations (60).

9. **Buckling of a Bar by Torsion and Flexure in an Elastic Medium.**—Let us consider the stability of a centrally compressed bar if, during buckling, there appear lateral reactions proportional to the deflection. We assume that these reactions are distributed along an axis N parallel to the axis of the bar and defined by the coördinates h_x and h_y, Fig. 27. Denoting, as before, the components of deflection of the shear-center axis by u and v and the angle of rotation with respect to that axis by φ, we find that the components of deflection of the N-axis, along which the reactions are distributed, are

$$u + (y_o - h_y)\varphi \quad \text{and} \quad v - (x_o - h_x)\varphi.$$

The corresponding reactions per unit length will be

$$-k_x[u + (y_o - h_y)\varphi] \quad \text{and} \quad -k_y[v - (x_o - h_x)\varphi],$$

* Equation (57) was developed from consideration of an element of the bar between two adjacent cross sections and is not affected by changes in the end conditions.

where k_x and k_y are constants defining the rigidity of the elastic medium. To these reactions we add lateral forces obtained from the action of initial compressive forces of the fibers on their slightly rotated cross sections. These previously discussed forces (see art. 8) give, per unit length,

$$-\int_A \sigma t ds[u'' + (y_o - y)\varphi''] \quad \text{and} \quad -\int_A \sigma t ds[v'' - (x_o - x)\varphi''].$$

Integrating and adding the results to the above calculated elastic reactions, we finally obtain the following expressions for the intensities of lateral force distribution:

$$\begin{aligned} q_x &= -k_x[u + (y_o - h_y)\varphi] - P(u'' + y_o\varphi''), \\ q_y &= -k_y[v - (x_o - h_x)\varphi] - P(v'' - x_o\varphi''). \end{aligned} \tag{66}$$

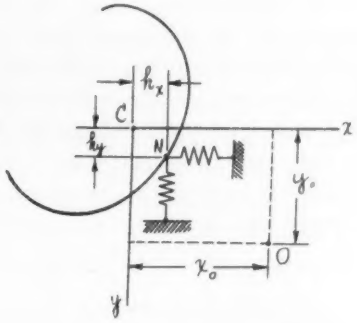

FIG. 27.

Substituting these into eqs. (18), we obtain

$$\begin{aligned} EI_y u^{IV} + P(u'' + y_o\varphi'') + k_x[u + (y_o - h_y)\varphi] &= 0, \\ EI_x v^{IV} + P(v'' - x_o\varphi'') + k_y[v - (x_o - h_x)\varphi] &= 0. \end{aligned} \tag{67}$$

To these equations of the deflection curve we have to add the equation for the angle of twist φ. Since the reactions are distributed along the axis N which does not coincide with the shear-center axis they will contribute to the torque distributed along the bar. The magnitude of this torque per unit length is

$$-k_x[u + (y_o - h_y)\varphi](y_o - h_y) + k_y[v - (x_o - h_x)\varphi](x_o - h_x).$$

The elastic medium may resist torsion also in the case where the N-axis coincides with the shear-center axis. This action on the bar can be represented by continuously distributed torque the intensity of which is proportional to the angle of twist and can be taken in the form $-k_\varphi\varphi$, where k_φ is again a constant depending on the rigidity of the medium. Adding these two components of the intensity of torque distribution to

that previously obtained from the action of initial compressive forces on rotated cross-sections of fibers (see art. 8), we obtain

$$m_z = (x_o v'' - y_o u'')P - r_o^2 P\varphi'' - k_x[u + (y_o - h_y)\varphi](y_o - h_y) + k_y[v - (x_o - h_x)\varphi](x_o - h_x) - k_\varphi \varphi. \quad (68)$$

With this value of m_z we obtain, instead of eq. (57) of the preceding article, the following equation for the angle of twist:

$$C_1\varphi^{IV} - (C - r_o^2 P)\varphi'' - P(x_o v'' - y_o u'') + k_x[u + (y_o - h_y)\varphi](y_o - h_y) - k_y[v - (x_o - h_x)\varphi](x_o - h_x) + k_\varphi \varphi = 0. \quad (69)$$

This equation, together with eqs. (67), defines the forms of buckling of the bar in an elastic medium and the corresponding values of the critical loads.*

If the ends of the bar cannot rotate but are free to warp, we take the solution of these equations in the form (63), and the calculation of critical loads will require, as before, the solution of a cubic equation. In some particular cases the calculation can be greatly simplified. Take, as a simple example, the bar with two planes of symmetry. In such a case, the shear-center axis coincides with the centroidal axis and we have $x_o = y_o = 0$. Assuming that elastic reactions are distributed along the centroidal axis, we have also $h_x = h_y = 0$ and eqs. (67), (69) become

$$EI_y u^{IV} + Pu'' + k_x u = 0,$$
$$EI_x v^{IV} + Pv'' + k_y v = 0,$$
$$C_1\varphi^{IV} - (C - r_o^2 P)\varphi'' + k_\varphi \varphi = 0.$$

We see that in this case buckling of the bar in the planes of symmetry is independent of torsion and the three forms of buckling can be discussed separately. The first two equations give the known critical loads for bending in the planes of symmetry. The last equation gives the critical load for torsional buckling. Taking the solution in the form (63), we find for the later critical load the expression

$$P_{cr} = \frac{C_1 \frac{n^2\pi^2}{l^2} + C + \frac{l^2}{n^2\pi^2} k_\varphi}{r_o^2}. \quad (70)$$

In each particular case, knowing C_1 and k_φ, we select for n the value which makes expression (70) a minimum. When $k_\varphi = 0$, we have to take $n = 1$ and the critical load (70) becomes equal to that previously obtained from eq. (55).

Consider now the case when the cross section of the bar has one axis of symmetry which we take as the x-axis. Then $y_o = o$. We as-

* These equations were first obtained by V. Z. Vlasov in his book, loc. cit., J. F. I., Apr. 1945, p. 256.

sume also that elastic reactions are distributed along the shear-center axis. Then $h_y = o$, $h_x = x_o$ and eqs. (67), (69) give

$$EI_y u^{IV} + Pu'' + k_x u = 0,$$

$$EI_x v^{IV} + Pv'' + k_y v - Px_o\varphi'' = 0,$$

$$C_1\varphi^{IV} - C\varphi'' + k_\varphi\varphi + Pr_o^2\varphi'' - Px_o v'' = 0.$$

We see that buckling in the plane of symmetry is independent of torsion and can be treated separately. Buckling in the direction perpendicular to the plane of symmetry is connected with torsion and we have to consider the last two of the above equations simultaneously. Taking the solution of these equations in the form (63), we obtain, for calculating the constants A_2 and A_3,

$$A_2\left(EI_x\frac{n^4\pi^4}{l^4} - P\frac{n^2\pi^2}{l^2} + k_\varphi\right) + Px_o\frac{n^2\pi^2}{l^2}A_3 = 0.$$

$$A_2Px_o\frac{n^2\pi^2}{l^2} + \left[C_1\frac{n^4\pi^4}{l^4} + (C - r_o^2P)\frac{n^2\pi^2}{l^2} + k_\varphi\right]A_3 = 0.$$

The critical value of the load is obtained by equating to zero the determinant of these two equations. In this way we obtain a quadratic equation for P_{cr} from which in each particular case the critical values of the load can be calculated. If k_φ vanishes, we have to take $n = 1$ and obtain for P_{cr} the equation which coincides with the previously derived eq. (62).

Using eqs. (67) and (69), we can investigate buckling of the bar in the case where the axis about which cross sections are rotating during buckling is prescribed. To obtain a rigid axis of rotation, we have only to assume that $k_x = k_y = \infty$. Then the axis N in Fig. 26 remains straight during buckling and cross sections rotate with respect to this axis. Equations (67) give in this case

$$u + (y_o - h_y)\varphi = 0,$$

$$v - (x_o - h_x)\varphi = 0.$$

and we obtain

$$u = -(y_o - h_y)\varphi, \qquad v = (x_o - h_x)\varphi.$$

We have also, from eqs. (67),

$$k_x[u + (y_o - h_y)\varphi] = -EI_y u^{IV} - P(u'' + y_o\varphi''),$$

$$k_y[v - (x_o - h_x)\varphi] = -EI_x v^{IV} - P(v'' - x_o\varphi'').$$

Substituting into eq. (69), we obtain

$$\varphi^{IV}[C_1 + (y_o - h_y)^2EI_y + (x_o - h_x)^2EI_x] - C\varphi'' + P\varphi''[r_o^2 - (x_o^2 + y_o^2) + h_x^2 + h_y^2] + k_\varphi\varphi = 0. \quad (71)$$

Taking the solution in the form (63), we can calculate in each particular case the critical load from this equation.

In the particular case of a bar with two planes of symmetry, we have $x_o = y_o = 0$. Substituting $\varphi = A_3 \sin \frac{n\pi z}{l}$ in eq. (71), we obtain

$$P_{cr} = \frac{(C_1 + h_y^2 EI_y + h_x^2 EI_x)\frac{n^2\pi^2}{l^2} + C + \frac{l^2}{n^2\pi^2}k_\varphi}{r_o^2 + h_x^2 + h_y^2}. \qquad (72)$$

In each particular case we have to take for n such value as to make expression (72) a minimum.

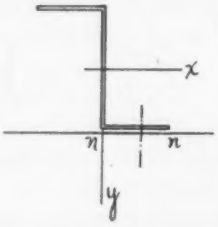

FIG. 28.

If the shear-center axis is taken as the axis of rotation, we have $h_x = x_o$, $h_y = y_o$, and eq. (71) gives

$$P_{cr} = \frac{C_1\frac{n^2\pi^2}{l^2} + C + \frac{l^2}{n^2\pi^2}k_\varphi}{r_o^2}. \qquad (73)$$

When k_φ vanishes, we have to take $n = 1$, and eq. (73) gives for P_{cr} the same value as eq. (55) as it should.

Taking the axis of rotation at infinite distance from the bar and assuming, for example, $h_y = \infty$, we obtain from eq. (71)

$$\varphi^{IV} EI_y + P\varphi'' = 0,$$

which gives for P_{cr} the known Euler load for buckling of the bar in the xz-plane.

Sometimes, instead of a fixed axis of rotation there may be prescribed a plane parallel to which certain fibers of the bar have to deflect during buckling. If a bar, for example, is welded to a thin sheet, Fig. 28, the fibers of the bar coinciding with the surface nn cannot deflect during buckling in the plane of the sheet but deflect in the direction perpendicular to that plane. In discussing such problems it is advantageous to take the centroidal axes x, y parallel and perpendicular to the sheet. Usually the axes will no longer be principal axes and the corresponding differential equations of the deflection curve must be changed. From

eqs. (1) and (2) we have, in the general case,

$$M_x = -EI_z v'' - EI_{xy} u'',$$
$$M_y = EI_y u'' + EI_{xy} v''.$$

Differentiating these equations twice with respect to z and using eqs. (5) and (6), we obtain

$$\left.\begin{aligned} q_y &= EI_z v^{IV} + EI_{xy} u^{IV}, \\ q_x &= EI_y u^{IV} + EI_{xy} v^{IV}. \end{aligned}\right\} \tag{74}$$

Considering any shape of cross section, Fig. 29, assume that a fiber N

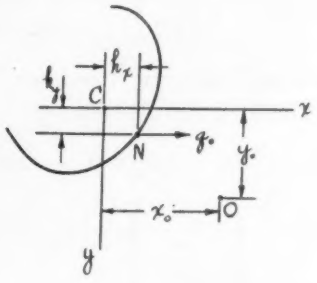

FIG. 29.

with coördinates h_x, h_y cannot deflect in the x-direction. Denoting, as before, by u and v deflections of shear-center axis O in the x- and y-directions, we find for deflections of the fiber N the following expressions:

$$u + \varphi(y_o - h_y), \qquad v - \varphi(x_o - h_x).$$

From the conditions of restraint of the fiber N we conclude that

$$u + \varphi(y_o - h_y) = 0. \tag{75}$$

Due to this restraint there will be reactions parallel to x-axis and continuously distributed along the fiber N. Let q_o be the intensity of this distributed force. Then the quantities q_x and q_y in eqs. (74) are obtained by substituting $k_x = k_y = 0$ in eqs. (66) and adding q_o to the first of these equations. In this manner we obtain

$$q_x = -P(u'' + y_o\varphi'') + q_o,$$
$$q_y = -P(v'' - x_o\varphi'').$$

Substituting into eqs. (74), we obtain

$$\begin{aligned} q_o &= P(u'' + y_o\varphi'') + EI_y u^{IV} + EI_{xy} v^{IV}, \\ EI_z v^{IV} &+ EI_{xy} u^{IV} + P(v'' - x_o\varphi'') = 0. \end{aligned} \tag{76}$$

The second of these equations, after substituting for u its value from

eq. (75), gives the following equation containing v and φ only:

$$EI_x v^{IV} + Pv'' - EI_{xy}(y_o - h_y)\varphi^{IV} - Px_o\varphi'' = 0. \tag{77}$$

The second equation for v and φ is obtained from a consideration of torsion of the bar. For this purpose we use eq. (69). Substituting in this equation $k_\varphi = k_y = 0$, $-k_x[u + (y_o - h_y)\varphi] = q_o$, we obtain

$$C_1\varphi^{IV} - (C - r_o^2 P)\varphi'' - P(x_o v'' - y_o u'') - q_o(y_o - h_y) = 0.$$

Substituting in this equation the value of u from eq. (75) and the value of q_o from the first of eqs. (76), we obtain

$$[C_1 + EI_y(y_o - h_y)^2]\varphi^{IV} - C\varphi'' + P\varphi''(r_o^2 + h_y^2 - y_o^2) - EI_{xy}(y_o - h_y)v^{IV} - Px_o v'' = 0. \tag{78}$$

Equations (77) and (78) will now be used for calculation of critical loads.* Assuming that the end conditions are

$$\varphi = \varphi'' = v = 0, \quad \text{for} \quad z = 0 \quad \text{and for} \quad z = l,$$

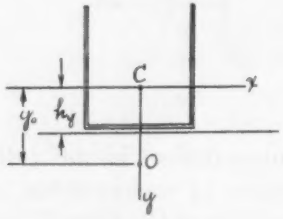

FIG. 30.

we take the solution of eqs. (77) and (78) in the form (58) and, after substitution, we obtain

$$\left(\frac{EI_x\pi^2}{l^2} - P\right)A_2 - \left[\frac{EI_{xy}\pi^2}{l^2}(y_o - h_y) - Px_o\right]A_3 = 0,$$

$$-\left[EI_{xy}(y_o - h_y)\frac{\pi^2}{l^2} - Px_o\right]A_2 + \left\{[C_1 + EI_y(y_o - h_y)^2]\frac{\pi^2}{l^2} + C - P(r_o^2 + h_y^2 - y_o^2)\right\}A_3 = 0.$$

Equating to zero the determinant of these equations we obtain, as before, a quadratic equation for P from which the critical load in each particular case can be calculated.

If the bar is symmetrical with respect to the y-axis, as in the case of a channel, Fig. 30, the x- and y-axes are principal axes; hence, $I_{xy} = 0$,

* These equations were obtained by J. N. Goodier, Bull., no. 27, 1941, Cornell University Engineering Experiment Station.

$x_o = 0$, and the above equations become

$$\left(\frac{EI_x\pi^2}{l^2} - P\right) A_2 = 0,$$

$$\left\{ [C_1 + EI_y(y_o - h_y)^2]\frac{\pi^2}{l^2} + C - P(r_o^2 + h_y^2 - y_o^2) \right\} A_3 = 0.$$

From the first of these equations, we obtain the Euler load for buckling in the plane of symmetry. From the second equation we obtain the torsional critical load, the axis of rotation being in the plane of the sheet,

$$P_{cr} = \frac{[C_1 + EI_y(y_o - h_y)^2]\frac{\pi^2}{l^2} + C}{r_o^2 + h_y^2 - y_o^2}. \qquad (79)$$

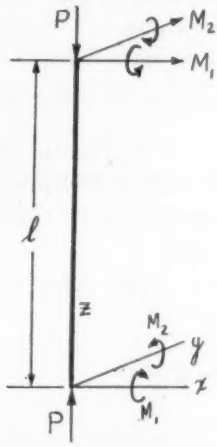

FIG. 31.

This same result would be obtained from eq. (71) by substituting $x_o = h_x = k_\varphi = 0$, $\varphi = A_3 \sin\frac{\pi z}{l}$.

10. **Stability of Thin-walled Members under Bending and Compression.**—It is known that beams bent in the plane of their maximum rigidity may fail due to 'ateral buckling accompanied by torsion.* Let us consider the case when the bar is submitted to the action of central thrust P and bending couples M_1 and M_2 applied at the ends, Fig. 31. We assume that the effect of P on bending stresses can be neglected. In

* This kind of buckling in the case of beams of a narrow rectangular cross section was investigated by L. Prandtl; see his dissertation, Nüremberg, 1899; and by A. G. M. Michell, *Phil. Mag.*, vol. 48, 1899. The case of an I-beam was discussed by the writer in his previously mentioned paper; see J. F. I., Apr. 1945, p. 254. The extension of the theory to bars of any thin-walled open section was made by V. Z. Vlasov in his previously mentioned book; see J. F. I., Apr. 1945, p. 256. Some cases are discussed also by J. N. Goodier, Bulletin no. 28, Engineering Experiment Station, Cornell University, 1942.

such case, the normal stress at any point is independent of z and is given by the equation

$$\sigma = -\frac{P}{A} + \frac{M_1 y}{I_x} - \frac{M_2 x}{I_y}, \tag{80}$$

in which x and y are the centroidal principal axes of the cross section. The initial deflection of the bar due to couples M_1 and M_2 we consider as very small. In investigating the stability of this deflected form of equilibrium, we proceed as before and assume that additional deflections u and v of the shear-center axis together with rotations φ with respect to that axis are produced, and write equations of equilibrium for this new form slightly different from the initial curved form produced by the couples M_1 and M_2. In writing these equations we neglect small initial deflections and proceed as in the preceding cases in which the axis of the bar was initially straight. Then the components of deflection of any longitudinal fiber of the bar, defined by coördinates x and y, are

$$u + (y_o - y)\varphi, \qquad v - (x_o - x)\varphi.$$

The intensities of the lateral fictitious loads and distributed torque obtained from the action of initial compressive forces in the fibers on their slightly rotated cross-sections are obtained as before (see art. 8) and are given by the equations

$$q_x = -\int_A \sigma t ds[u'' + (y_o - y)\varphi''],$$

$$q_y = -\int_A \sigma t ds[v'' - (x_o - x)\varphi''],$$

$$m_z = -\int_A \sigma t ds[u'' + (y_o - y)\varphi''](y_o - y) + \int_A \sigma t ds[v'' - (x_o - x)\varphi''](x_o - x).$$

Substituting expression (80) for σ and integrating, we obtain

$$q_x = -Pu'' - (Py_o + M_1)\varphi'',$$

$$q_y = -Pv'' + (Px_o - M_2)\varphi'',$$

$$m_z = -(Py_o + M_1)u'' + (Px_o - M_2)v'' + (M_1\beta_1 - M_2\beta_2 - Pr_o^2)\varphi'',$$

where the following notations are introduced:

$$\left.\begin{aligned} \beta_1 &= \frac{\int_A y^3 dA + \int_A x^2 y dA}{I_x} - 2y_o, \\ \beta_2 &= \frac{\int_A x^3 dA + \int_A x y^2 dA}{I_y} - 2x_o. \end{aligned}\right\} \tag{81}$$

Equations (18) and (51) then give

$$\left.\begin{aligned} EI_y u^{IV} + Pu'' + (Py_o + M_1)\varphi'' &= 0, \\ EI_x v^{IV} + Pv'' - (Px_o - M_2)\varphi'' &= 0, \\ C_1\varphi^{IV} - (C + M_1\beta_1 - M_2\beta_2 - Pr_o^2)\varphi'' & \\ + (Py_o + M_1)u'' - (Px_o - M_2)v'' &= 0. \end{aligned}\right\} \quad (82)$$

These are the general equations of equilibrium for the buckled form of the bar. From these equations the critical values of external forces can be calculated for any given end condition.

Let us begin with the case of eccentric thrust. If e_x and e_y denote the coördinates of the point of application of the thrust P, we have

$$M_1 = -Pe_y, \qquad M_2 = Pe_x.$$

Substituting in eqs. (82) we obtain

$$\left.\begin{aligned} EI_y u^{IV} + Pu'' + P(y_o - e_y)\varphi'' &= 0, \\ EI_x v^{IV} + Pv'' - P(x_o - e_x)\varphi'' &= 0, \\ C_1\varphi^{IV} - (C - Pe_y\beta_1 - Pe_x\beta_2 - Pr_o^2)\varphi'' & \\ + P(y_o - e_y)u'' - P(x_o - e_x)v'' &= 0. \end{aligned}\right\} \quad (83)$$

In the case of simply supported ends, we have the following end-conditions:

$$\left.\begin{aligned} u = v = \varphi &= 0, \\ u'' = v'' = \varphi'' &= 0, \end{aligned}\right\} \quad \text{for} \quad z = 0 \quad \text{and} \quad z = l.$$

We will satisfy these conditions by taking

$$u = A_1 \sin\frac{\pi z}{l}, \qquad v = A_2 \sin\frac{\pi z}{l}, \qquad \varphi = A_3 \sin\frac{\pi z}{l}.$$

Substituting into eqs. (83), we obtain

$$\left.\begin{aligned} \left(EI_y\frac{\pi^2}{l^2} - P\right)A_1 - P(y_o - e_y)A_3 &= 0, \\ \left(EI_x\frac{\pi^2}{l^2} - P\right)A_2 + P(x_o - e_x)A_3 &= 0, \\ -P(y_o - e_y)A_1 + P(x_o - e_x)A_2 & \\ + \left(C_1\frac{\pi^2}{l^2} + C - Pe_y\beta_1 - Pe_x\beta_2 - Pr_o^2\right)A_3 &= 0. \end{aligned}\right\} \quad (84)$$

Equating to zero the determinant of these equations, we obtain, as before, a cubic equation for calculating P_{cr}.

In a particular case when the thrust P acts along the shear-center axis, we have

$$e_x = x_o, \qquad e_y = y_o,$$

and eqs. (83) are greatly simplified, since each of them contains only one variable. Lateral buckling in two principal planes is independent of each other and independent of torsion. We obtain three critical values of P, two given by the Euler formula, and one corresponding to torsional buckling.

If the point of application of thrust does not coincide with the shear-center the three eqs. (83) are interconnected and flexure of the bar during buckling is connected with torsion.

The problem is considerably simplified if the bar has one plane of symmetry. Assume that the yz-plane is the plane of symmetry and that the thrust P acts in that plane. Then $x_o = e_x = 0$, and eqs. (84) become

$$\left.\begin{aligned} \left(EI_y\frac{\pi^2}{l^2} - P\right)A_1 - P(y_o - e_y)A_3 &= 0, \\ \left(EI_x\frac{\pi^2}{l^2} - P\right)A_2 &= 0, \\ -P(y_o - e_y)A_1 + \left(C_1\frac{\pi^2}{l^2} + C - Pe_y\beta_1 - Pr_o^2\right)A_3 &= 0. \end{aligned}\right\} \tag{85}$$

From the second equation we see that buckling in the plane of symmetry is independent of torsion and the corresponding critical load is the same as the Euler load. Buckling in the xz-plane and torsion are interconnected and the corresponding critical loads are obtained by equating to zero the determinant of the first and third of eqs. (85). This gives

$$\begin{vmatrix} \left(EI_y\frac{\pi^2}{l^2} - P\right), & -P(y_o - e_y), \\ -P(y_o - e_y), & \left(C_1\frac{\pi^2}{l^2} + C - Pe_y\beta_1 - Pr_o^2\right). \end{vmatrix} = 0$$

Using notations (60) and evaluating the determinant, we obtain for calculating critical loads the following quadratic equation:

$$(P_1 - P)[P_3r_o^2 - P(r_o^2 + e_y\beta_1)] - P^2(y_o - e_y)^2 = 0. \tag{86}$$

For very small values of P, the left-hand side of this equation is positive. When $P = P_1$, it is negative. Hence, there is a root of eq. (86) smaller than P_1, i.e. smaller than the Euler load for buckling in the xz-plane.

If $e_y = y_o$, the thrust is applied at the shear-center, and eq. (86) gives

$$(P_1 - P)[P_3r_o^2 - P(r_o^2 + e_y\beta_1)] = 0.$$

The two solutions of this equation are

$$P = P_1, \qquad P = \frac{P_3}{1 + \frac{e_y \beta_1}{r_o^2}}.$$

The first one corresponds to flexural buckling in the xz-plane; the second, to purely torsional buckling.

Since the left-hand side of eq. (86) is positive when P is small and gradually diminishes as P increases, we conclude that the smallest root of the equation will be increased by making the parenthesis in the last term vanish; i.e. by taking $e_y = y_o$. Hence the critical load reaches its maximum value if the thrust is applied at the shear center. If $e_y = 0$, eq. (86) is of the same kind as eq. (62) previously obtained for centrally applied thrust. If the cross section has two axes of symmetry, we have $y_o = \beta_1 = 0$ and eq. (86) becomes

$$(P_1 - P)(P_3 - P)r_o^2 - P^2 e_y^2 = 0,$$

or

$$P^2(r_o^2 - e_y^2) - P(P_3 + P_1)r_o^2 + P_1 P_3 r_o^2 = 0. \tag{87}$$

From this equation, we find

$$P = \frac{P_1 + P_3 \pm \sqrt{(P_3 - P_1)^2 + 4P_1 P_3 e_y^2 / r_o^2}}{2(1 - e_y^2/r_o^2)}. \tag{88}$$

It is seen that if $e_y < r_o$ there are two positive roots which approach the values P_1 and P_3 as e_y approaches zero. If $e_y > r_o$, there will be one root positive, another negative; the later indicating that for larger eccentricities, the bar may buckle under the action of eccentric tension. When $r_o = e_y$, we see from eq. (87) that one root is

$$P = \frac{P_1 P_3}{P_1 + P_3};$$

the other becomes infinity.

We discussed the case of a bar with simply supported ends. If the ends are built-in, the end conditions are

$$\left.\begin{aligned} u = v = \varphi = 0, \\ u' = v' = \varphi' = 0, \end{aligned}\right\} \quad \text{for} \quad z = 0 \quad \text{and} \quad z = l.$$

We will satisfy these conditions by taking

$$u = A_1\left(1 - \cos\frac{2\pi z}{l}\right), \qquad v = A_2\left(1 - \cos\frac{2\pi z}{l}\right),$$

$$\varphi = A_3\left(1 - \cos\frac{2\pi z}{l}\right).$$

Substituting in eqs. (83), we obtain, for calculation critical loads, an equation similar to that already derived for simply supported ends. The difference is only that $4\pi^2/l^2$ appears instead of π^2/l^2 which we had before.

Let us consider now the case when the bar is submitted to the action of pure bending. Substituting $P = 0$ in eqs. (82), we obtain

$$EI_y u^{IV} + M_1\varphi'' = 0,$$
$$EI_x v^{IV} + M_2\varphi'' = 0, \qquad (89)$$
$$C_1\varphi^{IV} - (C + M_1\beta_1 - M_2\beta_2)\varphi'' + M_1 u'' + M_2 v'' = 0.$$

Taking for u, v, and φ, the trigonometric expression which we used before, the equation for calculating critical values of bending moments can be readily derived.

Of particular interest is the case where the bar has flexural rigidity in one principal plane many times larger than in the other and is bent in the plane of greater rigidity. Assume, for example, that the yz-plane is the plane of greater rigidity and that the bar is bent in this plane by couples M_1. To determine the critical value of M_1 at which lateral buckling occurs, we use the first and last of eqs. (89). Assuming that the ends are simply supported, we substitute in these equations

$$u = A_1 \sin\frac{\pi z}{l}, \qquad \varphi = A_3 \sin\frac{\pi z}{l}, \qquad M_2 = 0.$$

and obtain

$$EI_y \frac{\pi^2}{l^2} A_1 - M_1 A_3 = 0,$$
$$-M_1 A_1 + \left(C_1\frac{\pi^2}{l^2} + C + M_1\beta_1\right) A_3 = 0.$$

Equating to zero the determinant of these equations and using notations (60), we obtain, for calculating the critical values of M_1, the equation

$$M_1^2 - P_1\beta_1 M_1 - P_2 P_3 r_o^2 = 0,$$

from which

$$(M_1)_{cr} = \frac{P_1\beta_1}{2} \pm \sqrt{\frac{P_1^2\beta_1^2}{4} + P_1 P_3 r_o^2}. \qquad (90)$$

In the particular case of a cross section with two axes of symmetry, β_1 vanishes and we obtain

$$(M_1)_{cr} = \sqrt{P_1 P_3 r_o^2} = \frac{\pi}{l}\sqrt{EI_y\left(C_1\frac{\pi^2}{l^2} + C\right)}. \qquad (91)$$

In the case of point symmetry, as in Fig. 32*a*, again $\beta_1 = 0$, and eq. (91) holds.

The same equation holds also in the case of cross sections with one axis of symmetry if the bending couple is acting in the plane perpendicular to the symmetry axis as in Fig. 32*b*, since β_1 vanishes also in this case. If the bending couple M_1 acts in the plane of symmetry, as in Fig. 32*c*, β_1 does not vanish, and eq. (90) must be used for the calculation of $(M_1)_{cr}$. It was assumed in the above discussion that EI_y is small in comparison with EI_x. If C and C_1 are also small, buckling will occur at small values of M_1, i.e. at small bending stresses in the bar.

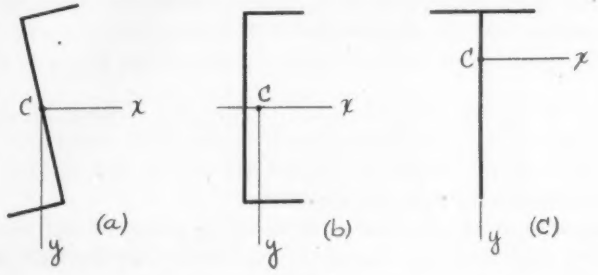

FIG. 32.

If EI_x is of the same order as EI_y, lateral buckling will occur at small stresses only if C and C_1 are very small. This condition may be fulfilled if the cross section is like that shown in Fig. 23. C_1 vanishes in this case and C is very small if the flange thickness is small.

We have discussed here bending of a bar by couples applied at the ends. Only in this case are the normal stresses (80) independent of z so that we obtain eqs. (82) with constant coefficients. If the bar is bent by lateral load, the bending stresses vary with z and, instead of eqs. (82), we will obtain a system of linear conditions with variable coefficients. The calculation of critical values of lateral load then becomes more involved. Several cases of this kind of instability have been discussed by the writer in his previously mentioned paper.*

* Loc. cit., J. F. I., Apr. 1945, p. 254. See also "Theory of Elastic Stability," p. 239.

"Electrical Thinker" Solves Mechanical Problems.—A new "electrical thinker" that gives answers to mechanical stress problems too complex for mathematical analysis by solving them on an electrical circuit was reported recently to the annual meeting of the American Institute of Electrical Engineers.

The apparatus, designed to meet new problems in the design and construction of turbo-generator units for the production of electricity, enables engineers to solve these problems, many of which had been considered "unsolvable," in a matter of minutes, said Dr. G. D. McCann, transmission engineer for the Westinghouse Electric and Manufacturing Company.

"Engineers have long recognized a relationship between mechanical and electrical systems, but the development of apparatus to carry through these relations for a practical purpose has been a stumbling block to their full use," he asserted.

The heart of the Westinghouse unit, called the mechanical-transient analyzer, includes specially designed low-loss electrical elements, particularly a new type of coil, which is used to produce the effects of a mechanical system in the electrical circuit, the speaker explained.

"In apparatus, such as turbo-generators, engineers must know what effect a short circuit will have on the shaft and other mechanical parts when the smooth and coordinated running of the apparatus is upset," Dr. McCann continued.

"Severe stresses occur when the orderly operation of the machinery is interrupted and the effect on parts, such as the shaft, is much the same as though a brake was suddenly applied to one of four wheels of a speeding car. If there is a breakdown in the equipment because it isn't built for peak stresses that may last for only a fraction of a second, then a power station is in for serious trouble.

"But with the newly developed analyzer, engineers can determine the worst conditions under which the apparatus might operate and carry out the design in line with such data."

The original analyzer, used as an experimental unit, is now being rebuilt and will be housed in a small laboratory room. The setting up of a problem on it merely involves the connection of the necessary electrical circuits to represent the mechanical system, the Westinghouse engineer said.

For example, he continued, an inductance coil is used to represent the mass in a mechanical system such as a turbine and generator, while the force involved in the rotation of the equipment is represented by voltage applied to the analyzer circuit.

"From the results of the electrical test we get an equation," Dr. McCann added, "and it is applied to the problem of the mechanical system under test. Without the analyzer, many problems of determining the peak stresses of turbine-generator shafts would be far too complex for ordinary mathematical solution."

Although the mechanical-transient analyzer was developed primarily for meeting problems involved in power station equipment, it can be used for almost any mechanical stress problem, Dr. McCann stated. These include, he added, the determination of peak stresses in an airplane at the onset of "flutter" when a plane reaches the speed of sound from 740 to 760 miles per hour, and the "ridability" of an automobile.

R. H. O.

RADIO ANTENNA SUSPENDED FROM 1,000 FOOT TOWERS.

BY

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INTRODUCTION.

The years immediately following World War I saw the development of many high radio antenna and tower stations. A similar program of long distance transmission requirements should follow the close of the present war. A detailed description of a study made in checking the design of 1,000 ft. guyed radio towers and in designing mat antenna 2,800 ft. square, with basic data brought up to date, should be of use in the design of such structures. At the time of design and construction, these were among the highest steel structures.

GENERAL DESCRIPTION OF THE PROJECT.

As a central station located near Shanghai, the Federal Telegraph Company planned a group of seven towers, arranged in two squares with one common tower. Each square, 2,800 feet on a side, had an independenent antenna consisting of 8 concentric squares. Diagonal halyards connected the antenna squares and were supported by the towers. The controlling dimension, in addition to the length of the antenna wires, was the clear height of the antenna above the ground. A "rat-tail" connecting cable ran from the halyard to the ground. Towers are 6 feet square, 1,000 feet in total height, guyed at eight approximately equidistant points by cable guys in four directions. Towers and antenna were to be insulated as determined necessary by the electrical requirements. The project was designed for future extension.

The report is divided into the following sections:

A. Discussion on Wind Pressure.
B. Station Layout.
C. Design of Antenna.
D. Design of Guys for Towers.
E. Design of Towers.

A. DISCUSSION ON WIND PRESSURE.

For tall structures designed to resist wind, the following data are needed:

1. Maximum wind velocity to be considered.
2. Extent of maximum wind velocity.

3. Relation between velocity and height above ground.
4. Corresponding pressure of wind on tower and on cables.
5. Direction of wind and effect of inclination on pressure.

Maximum Wind Velocity.

Available records indicate large variation in opinion regarding the velocities to be assumed in designing high structures. From the best available information, an actual velocity of 75 miles per hour at ground level and 150 miles per hour at an elevation of 1,000 feet, seems to be the maximum to be a reasonable expectation almost anywhere in the world.

In considering reported velocities which are to be converted into pressures acting, care must be taken in determining how the velocities were measured. In the United States the Weather Bureau reports up to 1928 were the result of readings from 4 cup Robinson anemometers. From 1928 to 1931, inclusive, the readings were made by 3 cup anemometers. Since 1932, true velocities are reported. Fig. 1 shows the relation between velocities measured by 4 cup and 3 cup anemometers with true velocities. Approximately, the 3 cup reading is 5 per cent. above true velocity and the 4 cup reading is 30 per cent. too high.

Radio stations are normally located in open country where full exposure is the rule and there are no shielding structures. The design must take into account the pressure of the maximum wind with localized gust effects. The maximum wind in any locality has probably not yet occurred nor is there a record of its velocity. Available wind velocity records indicate a probable range for design; the factor of safety used in the choice of allowable maximum stresses should cover excess values. It is not economical to design for reported or assumed tornado velocities; even in tornado country, the coincidence of storm center and radio station location is unlikely.

High velocities recorded are listed in Table 1. These are true velocities; the proper correction having been made for the method of measuring. It will be noted that the highest velocities occur at high altitudes, that 130 m.p.h. is about the maximum at sea level and that wind velocities above 100 have been seldom recorded.

Tornado wind velocities at the localized storm center are probably in the vicinity of 200 m.p.h. The failure of water tanks in the 1925 tornado in Southern Illinois indicated that velocity. Damage during the 1926 tornado at Charles Co., Md., indicated at least a velocity of 150 m.p.h.

S. P. Wing, M. Am. Soc. C. E. (1), reports velocity records of 90 m.p.h. in New York in 1912, 1914, and 1915, and recommends an increase of 25 per cent. over anemometer readings for design consideration of gusts.

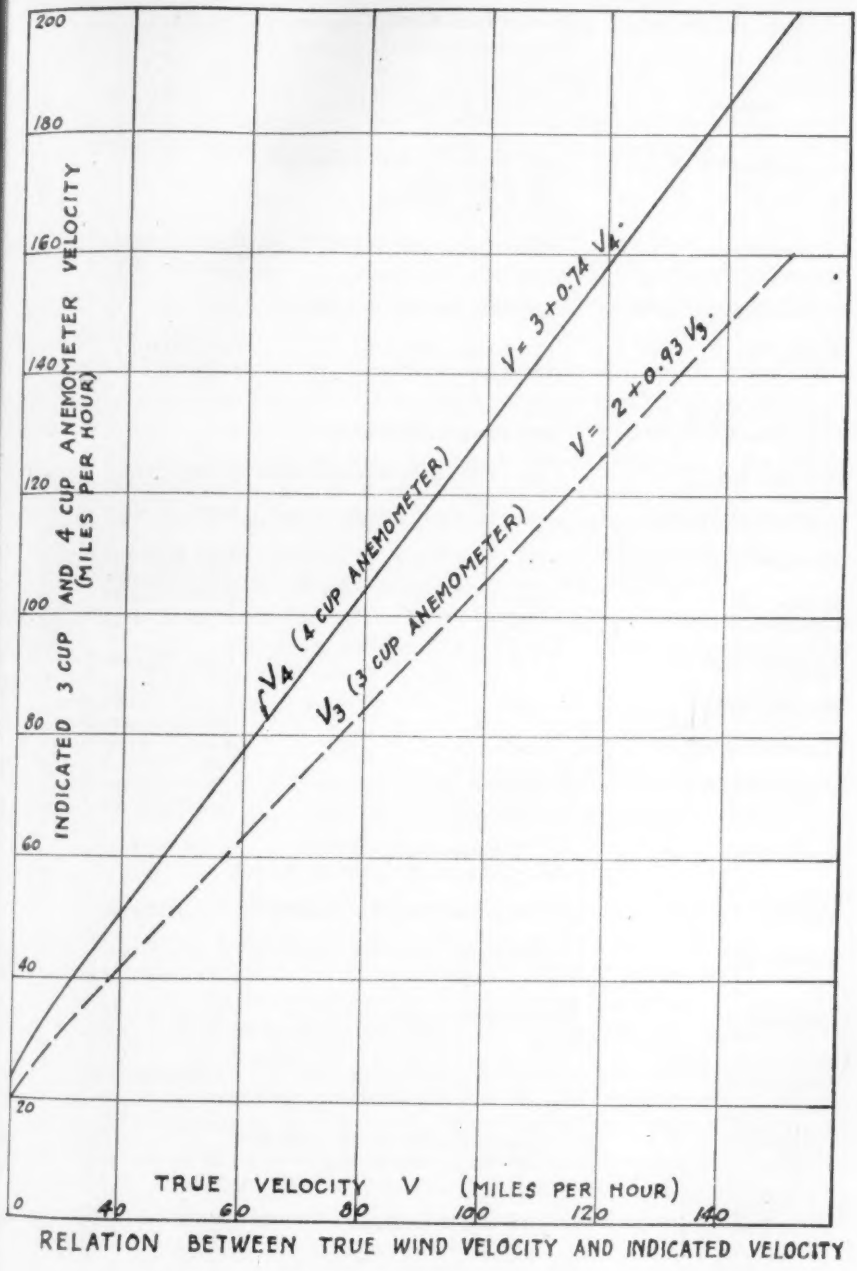

200
180
160
140
120
100
80
60
40
20
0
INDICATED 3 CUP AND 4 CUP ANEMOMETER VELOCITY (MILES PER HOUR)
V_4 (4 CUP ANEMOMETER)
V_3 (3 CUP ANEMOMETER)
$V = 3 + 0.74\ V_4$.
$V = 2 + 0.93\ V_3$.
40
60
80
100
120
140
TRUE VELOCITY V (MILES PER HOUR)
RELATION BETWEEN TRUE WIND VELOCITY AND INDICATED VELOCITY

FIG. 1.

TABLE 1.

Recorded High Wind Velocities.
(converted to actual velocity)

Location	Velocity Recorded (M.p.h.)	Reference	Date
U. S. A.:			
Mt. Washington, N. H.	231 fastest 0.1 mile actual vel. 225 fastest mile 188 5 min. average		
Mt. Tamalpais, Calif.	148 extreme gusts	Weather Bureau, San Francisco	1915
Mt. Washington, N. H.	140 186 on 4 cup anemometer		1876
Miami, Fla.	132 2 min. actual vel. 122 5 min. average	Eng. N. Rec. 12–9–26	1926
Cape Lookout, N. C.	105		1879
West Palm Beach, Fla.	100 4 cup anemometer		1926
St. Louis, Mo.	96 3 miles from tornado center	ENR 9–29–27	1927
North Head, Wash.	96 114 m.p.h. for max. 1 min.	ENR 12–9–26	1921
Burrwood, La.	94	First Progress Rep., ASCE Comm. No. 31	1915
Mobile, Ala.	88	ENR 12–9–26	1916
Pensacola, Fla.	87	ENR 12–9–26	1916
Tatoosh, Wash.	84	First Progress Rep., ASCE Comm. No. 31	1921
Manteo, N. C.	80	First Progress Rep., ASCE Comm. No. 31	1879
Hatteras, N. C.	80	First Progress Rep., ASCE Comm. No. 31	1899
Cape Henry, Va.	80	First Progress Rep., ASCE Comm. No. 31	1899
Lorain, Ohio	77 1 min. max. velocity	ENR 6–28–24	1924
Europe:			
England	110 site of exceptional exposure	T. E. Stanton	1924
Continental	100 max., recording 1925 to 1933	O. Flachsbart Civ. Eng., August	1933 1933
England	85–100 gusts—very rarely	D. H. Remfrey	1922
Asia:			
Philippines	75 60 min. average 90 max. gusts with 4 to 8 sec. cycle	Zikawei Obser.	1923
Hongkong	130 estimated during typhoon		1922
India	100 max. gusts to 165	D. H. Remfrey	
West Indies:			
Porto Rico	160 1 min. average 150 5 min. average	ENR 11–29–28	1928
Swan Island	130–137 estimated in hurricane	A. W. Buel	1923
Cape San Antonio	120–140 estimated in hurricane	Inst. Radio Eng.	

Extent of Maximum Velocity.

Wind velocity is not steady, except in very light winds, and maximums occur as a series of gusts. Winds with average 1 mile velocity of 40 to 50 m.p.h. may have gusts 30 per cent. higher, as was demonstrated in Dines pressure tube anemometer tests in Washington, D. C. Patterson (2) recommends the formula for velocity of gusts, V_t:

$$V_t = 3 + 1.64 V_m, \tag{1}$$

where V_m is mean hourly velocity.

W. Walters Pagon (3) describes the general localized character of maximum velocities and the effect of shape of structure on acting wind velocity.

K. S. Sherlock and M. B. Stout (4) measured simultaneously the wind velocity at the top of 50 ft. poles spaced 60 feet apart covering a line 420 feet long. A variation of 50 per cent. (of the lower reading) was not uncommon in gusty winds and sometimes there was a 50 per cent. variation (26 to 39 m.p.h.) in readings only 60 feet apart. Measurements were also made on 5 points, 50 feet apart vertically, on a 250 ft. tower. In 3 sets of successive readings, recorded automatically every $\frac{1}{2}$ second, during gusty winds of 40 to 50 m.p.h., covering a total time of 70 seconds, the maximum velocity was recorded more often at the 150 and 200 ft. levels than at the 250 ft. level. The record of readings shows:

Max. vel. at elev.	50	100	150	200	250
No. of reading sets	20	17	38	38	27

In gusts, a 10 m.p.h. variation in 50 feet of height is common. Only in steady winds did the velocity always increase with the height.

Relation Between Velocity and Height Above Ground.

It is generally conceded that wind velocities at ground level are lower than above ground, but the differences and variation at any time are not exactly known. It should be noted that equal wind velocities at sea level near the shore and at an elevation of 1,000 feet inland do not require equal resistances for balancing the acting force. Moisture content as well as air density is a factor which must be considered.

The older theories of velocity variation with height are quoted by Dr. W. J. Humphreys in his "Physics of the Air" (5). The earliest formula, that of Stevenson, was based on measurements made in 1880, not exceeding 50 feet above the ground, and is hardly applicable to radio towers. Experiments made by E. Douglas Archibald in 1883–5 (6) with anemometers suspended from kites up to 1,300 feet above the ground, which was about 500 feet above sea level, resulted in a formula:

$$\frac{V}{v} = \left(\frac{H}{h}\right)^x, \tag{2}$$

and Archibald recommends a value of 0.25 for x. Velocities V and v occur at heights H and h above the ground. The average experimental values for x are shown in Table 2.

An approximation is given in the Weather Bureau publications in 1922 as follows: "The velocity at 500 meters above the surface is approximately double that at the surface and the increase is nearly linear with the height" (7).

The most valuable observations on the subject were made by S. P. Wing at Kerry, Ireland, in 1915 (8). Readings were made with 3 cup anemometers, two at fixed locations on top of 492 ft. and 300 ft. towers and the third at varying heights above ground. Wind velocities recorded were up to 37 m.p.h., with greater values at the higher loca-

TABLE 2.

Archibald Measurements of Wind Velocities at Various Elevations.

Test Series	Date	Mean Height (ft.) of Instrument Above Ground		Mean Wind Velocity (m.p.h.) at Instrument		Values of "x"
		Upper	Lower	Upper	Lower	
1	1883–4	250	102	18.4	13.4	0.372
2	1883–4	322	128	25.4	19.1	0.307
3	1883–4	407	179	19.4	15.8	0.275
4	1885–6	549	252	24.0	20.2	0.237
5	1885–6	795	481	25.0	22.2	0.252
6	1885–6	1095	767	25.4	23.8	0.194

Archibald recommends a value of 0.250 for "x" as a correction for ground elevation of the site of the tests.

tions, and with greater variations in individual pairs of measurements. He derived the formula:

$$V = 0.5[1 + \sqrt{(1 + 0.0047h)(1 - 0.000265h)}]V_g, \quad (3)$$

$$P = (0.00126h + 1.16)P_g, \quad (4)$$

in which V and P represent the velocity (m.p.h.) and pressure (lbs. per sq. ft.) at elevation h (ft.) and V_g and P_g are the corresponding values at the ground.

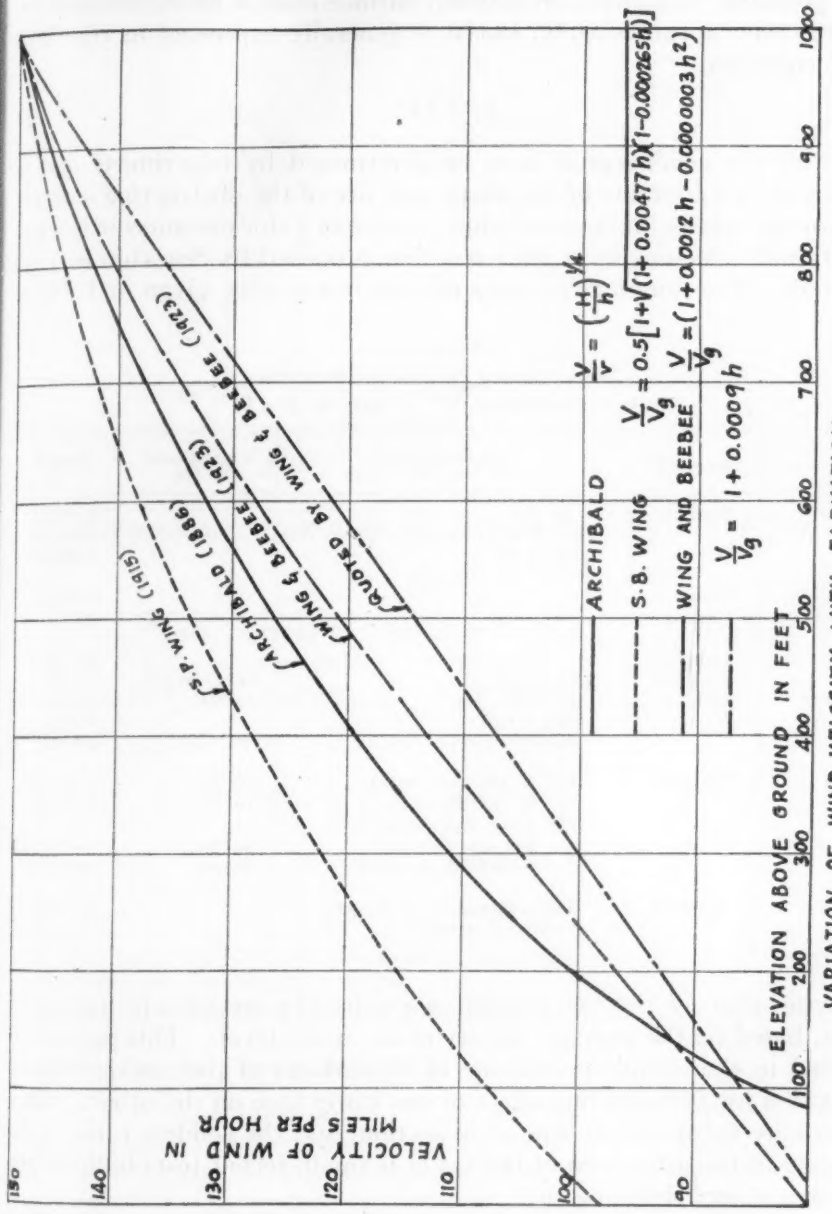
Chas. B. Wing and Ralph Beebee, Members A.S.C.E., in 1923 recommended two formulas:

$$V = (1 + 0.0009H)V_g, \quad (5)$$

$$V = (1 + 0.0012H - 0.0000003H^2)V_g, \quad (6)$$

but placed preference on formula (6).

Formulas (2), (3), (5) and (6) are plotted in Fig. 2, and show but little variation in results, when plotted through a velocity of 150 m.p.h. at 1000 ft. elevation.



VARIATION OF WIND VELOCITY WITH ELEVATION

FIG. 2.

Corresponding Pressure of Wind on Tower and on Cables.

The relation between velocity (V) of an air current and the resultant unit pressure (p) against an exposed surface cannot be determined by theory, even approximately, but it is generally expressed in the form of an equation:

$$p = kV^2, \tag{7}$$

in which the coefficient k is to be determined by experiment and is known to be a function of the shape and size of the obstructing surface.

For flat plates and normal wind, a value of k (for pressures in lbs. per sq. ft. and velocities in m.p.h.) was first proposed by Smeaton in 1759 as 0.005. The summary of more reliable test results, given in Table 3,

TABLE 3.

Tests of Normal Pressure on Flat Plates.

Values of coefficient "k" in equation: $p = kV^2$.

Reference	Reported by	Shape and Size of Test Area	Wind Velocity at Test	Value of "k"
(10)	Stanton	5′ × 5′		0.0032
		10′ × 5′		0.00318
		10′ × 10′		0.00322
	Dines			0.0029
	Froude			0.0036
	Langley			0.0032
	Chernet	8″ × 4½″	12–27 m.p.h.	0.0033
(11)	Eiffel	4″ to 40″ sq.	20–80	0.0033
	Pagon	long strips		0.0051
		square areas		0.0033
(17)	Durand	1¼″ × .05″ flat metal	50–70	0.004108
		1¼″ × .20″ flat wood	50–70	0.004324
		1¼″ diam. half round:		
		flat side to wind	50–70	0.004474
		round side to wind	50–70	0.002818
(19)	Stanton	Truss frames on gross area		0.00275
		Center of panels		0.00326

indicates that for the tower surfaces a value of 0.0033 is a proper coefficient, based on the average density of air at sea level. This pressure is applied to the complete exposure of all surfaces of the tower, with no reduction for the shielding effect of one tower face on the other. Since towers are substantially square in section, and the solidity ratio of the framing to the gross area of the tower is small, recent tests indicate the absence of any shielding (9).

For wires or cables, the normal pressure coefficient is less than for the case of flat surfaces. Theoretically, the ratio between cylindrical and flat surface resistances should be $\frac{1}{4}\pi$ or 0.7854. The list of reliable

tests on cables (see Table 4) shows almost an exact agreement between the empirical data and the value of 0.7854 × 0.0033 or 0.0026.

The theoretical coefficient for cylinders is determined as follows (see Fig. 3(a)):

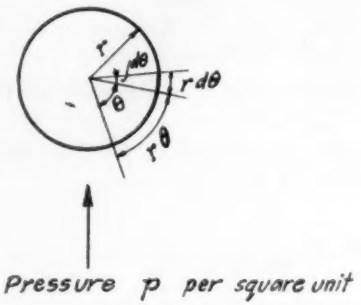


FIG. 3 (a)

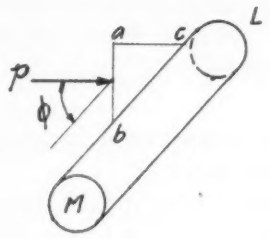


FIG. 3 (b)

Theoretical Formulas for Wind Pressure on Wires

FIG. 3.

Projected area normal to wind $= r\,d\theta \cos\theta$
Normal force on element $= p \cos\theta$
Vertical force on element $= pr \cos^2\theta\, d\theta$

Total vertical force on cylinder is

$$P = 2\int_0^{\frac{1}{2}\pi} pr\cos^2\theta\, d\theta$$
$$= 2pr(\tfrac{1}{2}\theta + \sin\theta\cos\theta)_0^{\frac{1}{2}\pi}$$
$$= 2pr\frac{\pi}{4} = \frac{1}{4}\pi pd. \qquad (8)$$

Pressure on flat surface $= pd$.
Ratio of pressures on cylinder to that on flat surfaces $= \frac{1}{4}\pi$.

Resultant Pressures under Inclined Wind Loads.

The wind is assumed normal to the mast or tower surfaces, but the guys are inclined to the assumed horizontal wind direction. The usual conversion formula between unit wind pressure (p) and its normal component on a sloping surface (p_n), where ϕ is the angle of inclination, is the Duchemin (20) formula:

$$P_n = p\frac{2\sin\phi}{1 + \sin^2\phi}. \qquad (9)$$

Even for flat plates, the validity of this formula is open to question. While Langley (18) found it in close agreement with his well known series of experiments, a comparison with the experiments of Lilienthal, Dines, Foeppl, Eiffel and others shows a wide divergence between tests and the values obtained by the formula (9).

For circular wires and cables, the theoretical solution, developed below, gives:

$$P_n = p \sin^2 \phi, \tag{10}$$

and the several tests listed in Table 5 agree quite well with each other and formula (10).

In Fig. 3(b), a horizontal wind of intensity, p, acts on a wire LM inclined at angle ϕ to the direction of p.

$$bc = \frac{ab}{\sin \phi}$$

unit wind pressure on bc is: $p \sin \phi$,
component normal to bc is: $p \sin^2 \phi$.

Wind Loads Used in Design of Some Existing Structures and Recommended.

(1) R.C.A. Port Jefferson (Long Island, N. Y.).
30 lbs. per sq. ft. of vertical surface on twice the projected area for upper 300 feet and on 1½ times the projected area for lower part of tower.

(2) U. S. Navy Towers (1920 U. S. Naval Inst. Proc.).
30 lbs. per sq. ft. on 1½ times the exposed area.

(3) A. W. Buel (21) (recommended in hurricane belt).
80 lbs. per sq. ft. on antenna.
68 lbs. on 1½ times the projected area on top story.
43.2 lbs. per sq. ft. on entire tower.

(4) Italian and Naval Station at Rome (714 ft. mast).
61.5 lbs. per sq. ft. on projected area.

(5) Eiffel Tower (Two assumptions were used).
(a) 61.4 lbs. per sq. ft. on whole tower
(b) varied from 40.9 at base to 81.9 per sq. ft. at top of tower.

(6) Radio Tower at Nashville, Tenn. (878 ft. mast).
30 lbs. per sq. ft. on projected area.

(7) WGY Tower at Schenectady, N. Y. (625 ft. guyed).
25 lbs. per sq. ft. for 300 feet and 35 lbs. above, on 150 per cent. of projected area, plus ½″ ice covering on entire structure.

(8) Power line over Puget Sound, Wash. (6,153 ft. span).
72 m.p.h. wind plus ½″ ice covering.

(9) National Electrical Safety Code, loading conditions for transmission lines.

(a) Light District—12 lbs. per sq. ft. of projected wire diameter.
(b) Medium District—8 lbs. per sq. ft. on projected wire diameter increased by a $\frac{1}{4}$" ice covering.
(c) Heavy District—8 lbs. per sq. ft. on projected wire diameter increased by a 1" ice covering.

TABLE 4.

Tests on Normal Pressure on Wires and Stranded Cables.
Values of coefficient "k" in equation: $p = kV^2$.

Reference	Reported by	Shape and Size of Test Area	Wind Velocity at Test	Value of "k"
(11)	Eiffel	.068, .088, .111 and .147" cables	9–68 m.p.h.	0.0027
		.04–.08" wires	9–68	0.00237
		1" smooth wire	9–68	0.002194
(12)		.146" cable	9	0.00237
		.146" cable	30	0.00256
		.146" cable	68	0.00267
(11)	Prandtl	.13–.30" cables		0.0027
(10)	Stanton			0.00225
(13)	Buck	.58" steel cable	8–33	0.0025
(14)	Relf and Powell	$\frac{3}{8}$" wire	27.3	0.002405
		$\frac{3}{8}$, $\frac{7}{16}$ and $\frac{1}{2}$" cables	27.3	0.00251
(15)	McLeod	.19 steel cable		0.002032
(16)	Foeppl	$\frac{1}{8}$–.3" wire cable	10–22	0.00274
		.27–.4" hemp cable	10–22	0.00294
		.17" hemp rope	10–22	0.00229
		.63" 4 strand hemp	10–18	0.00268
(17)	Durand	$1\frac{1}{4}$" rope lay steel	50–70	0.002366
		$1\frac{1}{4}$" wire lay steel	50–70	0.002194
		$\frac{3}{4}$" rope lay steel	50–70	0.002388
		$\frac{3}{4}$" wire lay steel	50–70	0.002517
		$\frac{5}{16}$–$\frac{3}{8}$" wire lay	50–70	0.0029

Recommended Loadings for Design.

Except in unusual exposures, where recorded wind velocities are considerably greater than noted below, the following design loads are recommended:

(*a*) A unit pressure of 20 lbs. per sq. ft. on exposed flat surfaces at ground level, increasing at the rate of 5 lbs. for each 100 feet of height. This is equivalent to a wind velocity range from ground to elevation 1,000 feet of 77 to 160 m.p.h. (if the conversion factor is 0.0033) or of 67 to 140 m.p.h. (if the conversion factor is 0.0044 as seems likely from tests on long strip areas similar to the legs of a tower). The wind velocities listed above take into account reduced air density with height.

(*b*) Pressure on tower surfaces shall be taken as horizontal and acting on all exposed surfaces with no reduction for shielding.

(*c*) Pressure on cables of 20 lbs. per sq. ft. of the projected area (length times diameter) at ground level, increasing at the rate of 3 lbs.

for each 100 feet of height. This is equivalent to a wind velocity from ground to elevation 1,000 feet of 90 to 145 m.p.h.

(*d*) Where cables are not substantially horizontal, the wind pressure is multiplied by $\sin^2 \phi$, since only the normal component of the wind pressure affects the cable stresses.

(*e*) For cables, the wind pressure load shall be that equivalent to the average elevation of each cable.

B. STATION LAYOUT.

A unit of the radio station consists of four 1,000 ft. high 6 ft. square, guyed towers, in a square array 2,800 feet on centers, supporting a wire antenna consisting of eight concentric squares. The antenna wires are joined by diagonal halyard ropes, which pass over 5 ft. diameter sheaves to a system of deflecting sheaves attached to the towers and to a counterweight system located at the ground. Insulators are introduced in the

TABLE 5.

Tests on Wires and Cables Inclined to Wind Direction.

Reference	Shape of Test Area	Wind Vel.	Values of "$p_n : p$" for values of ϕ							
			10°	20°	30°	40°	45°	50°	60°	70°
(11) Eiffel	.06″ wire	—			.20		.43		.65	
(11) Eiffel	.15″ cable	11 to 68			.195		.435		.71	
(14) Relf	$\frac{3}{8}$–$\frac{1}{2}$″	27	.035	.12	.255	.40		.58	.75	.90
(17) Durand	$1\frac{1}{4}$″	50–70	.043	.125	.25	.40		.57	.75	.88
Values of $\sin^2 \phi$:			.03	.12	.25	.41	.50	.59	.75	.88

antenna spans, in the halyard rope and in the tower as required by electrical design requirements.

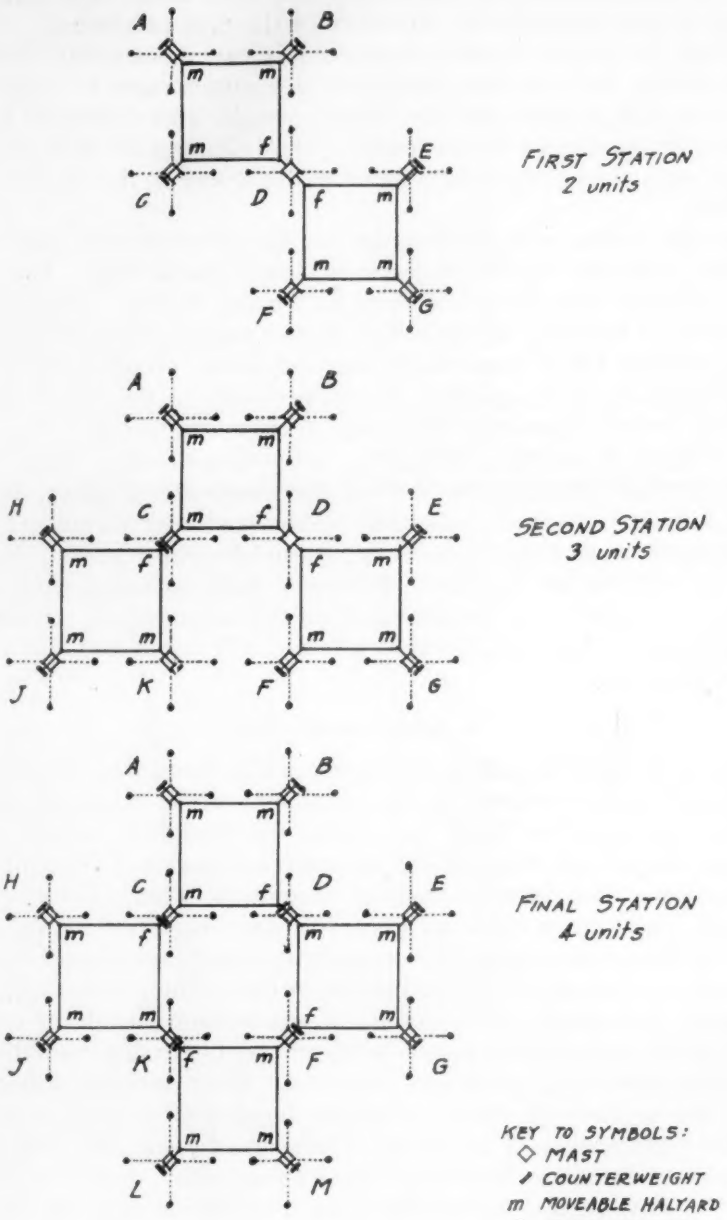
The use of counterweights fixed the maximum tensions in the halyards and in turn controlled the average sag of the antenna wires. Counterweighting three corners of the antenna mat is sufficient control, the fourth halyard being fixed to the tower. The resultant pull on the tower however is most severe for the fixed condition. It was planned to expand the station, starting with two antenna units, to three and later four units, as is shown in Fig. 4(a), (b) and (c).

There were three requirements on the plan in each stage:

1. Each unit must have three moveable halyards.
2. No outside mast shall have a fixed halyard.
3. Only one moveable halyard at any mast.

The first station of two units required seven masts, six with counterweights and a center common mast to which one halyard of each antenna is fixed.

To add a third unit antenna requires three more masts, each with halyard counterweight. The fourth halyard is fixed to a tower which



PLAN FOR STATION EXTENSION

FIG. 4.

carries a counterweighted halyard of the first unit. The additional work in no way disturbs the operation of the first two units.

To add the fourth antenna requires only two more masts, each with counterweight, but also the addition of a counterweight to center mast of the two-unit station and the counterweight and deflecting sheaves of one other mast must be reversed. These changes all affect the same antenna unit, so that two units can remain in service during the alteration work.

Antenna masts must be designed for the severest load combination resulting from any of the stages in station construction. Deflections of the antenna control its efficiency as a radio station. Sags must be controlled by limiting the deflection of the masts under wind load and by introducing initial tensions in antenna wires. Refinement of theoretical solutions of the problem is justified only to the extent that the resulting accuracy is within the range of certainty of exposed loadings. Dead weights of antenna, insulators, halyards and other items making up the antenna systems, as well as the mast and its guys, are quite closely known. Accurate formulas for stress determinations, deflections and initial tensions with the resulting geometric shapes under "No Wind Load" conditions are justified. However under assumed wind and ice conditions, similar time-consuming detailed computations are not justified and approximations, carefully chosen, will not reduce the correctness of the results.

C. DESIGN OF ANTENNA.

The makeup of the antenna is shown in Fig. 5 and the characteristics, weight and wind exposure, of the component wires, insulators and connections are listed in Table 6. Under no wind load conditions, the antenna shape and stresses are symmetrical and not difficult of determination. The introduction of insulators, connections and the "rat-tail" lead-down cable weights, forming discontinuous rigid loads in the flexible cables, somewhat complicates and makes uncertain both the stress and the shape determination in the vicinity of the rigid loads. However, from static requirements of equilibrium, resultant reactions at any point and average slopes between points are determinable.

Under dead load conditions, the seven inner antenna wires, $\frac{15}{32}$ in. diam., are to have an initial minimum tension of 5,000 lbs. each; the outer or eighth wires, $\frac{5}{8}$ in. diam., a tension of 7,500 lbs. These were electrical requirements for proper vibration periods. Stresses and shape of the wires were determined from the assumption that: 1. Spans are horizontal, because of the symmetry of the antenna. 2. Curves are parabolic, which is closely correct for the low sag to span ratios and under the initial tension requirements. A rigid solution involves the use of the "elastic catenary" which solution has been published elsewhere (22).

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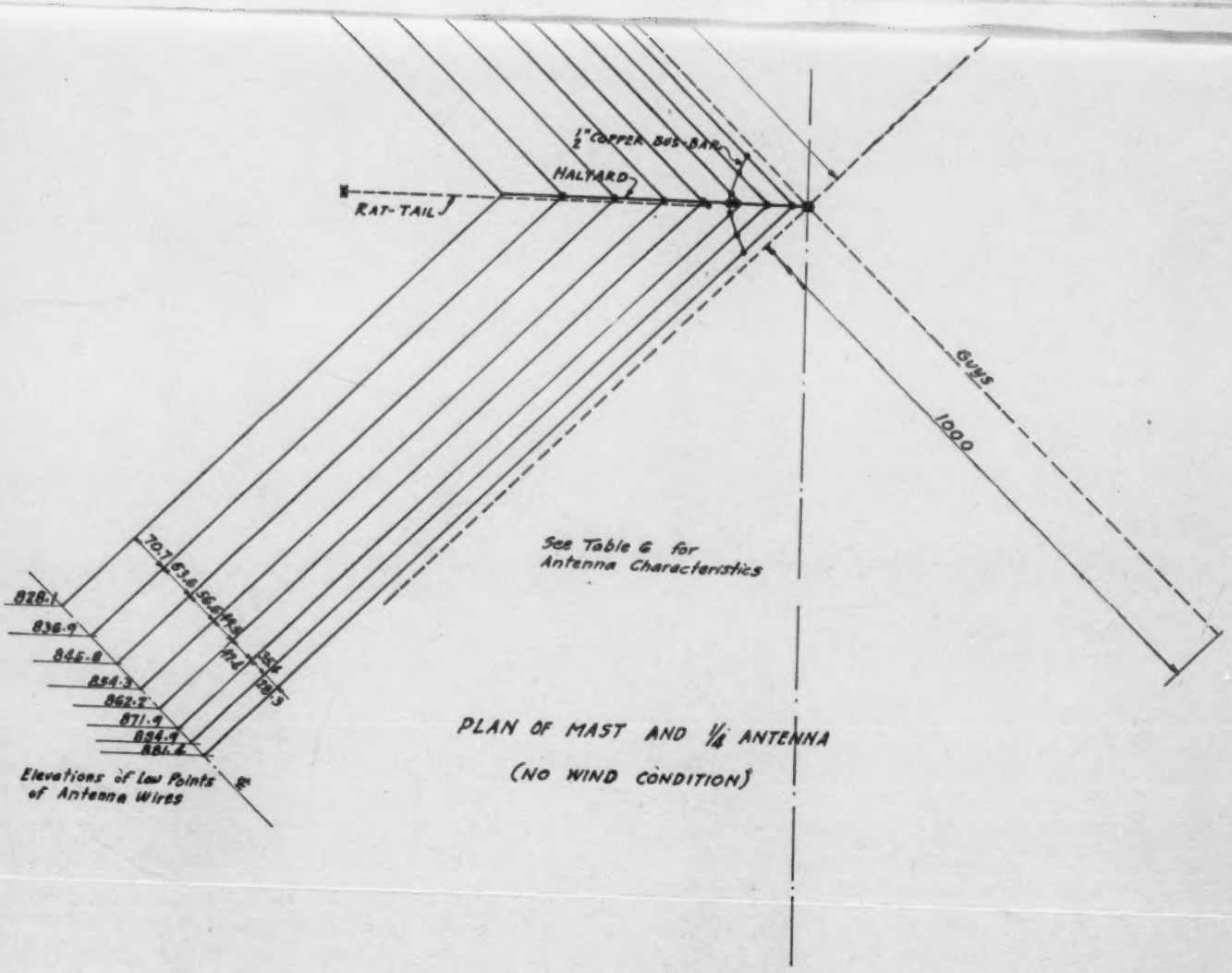
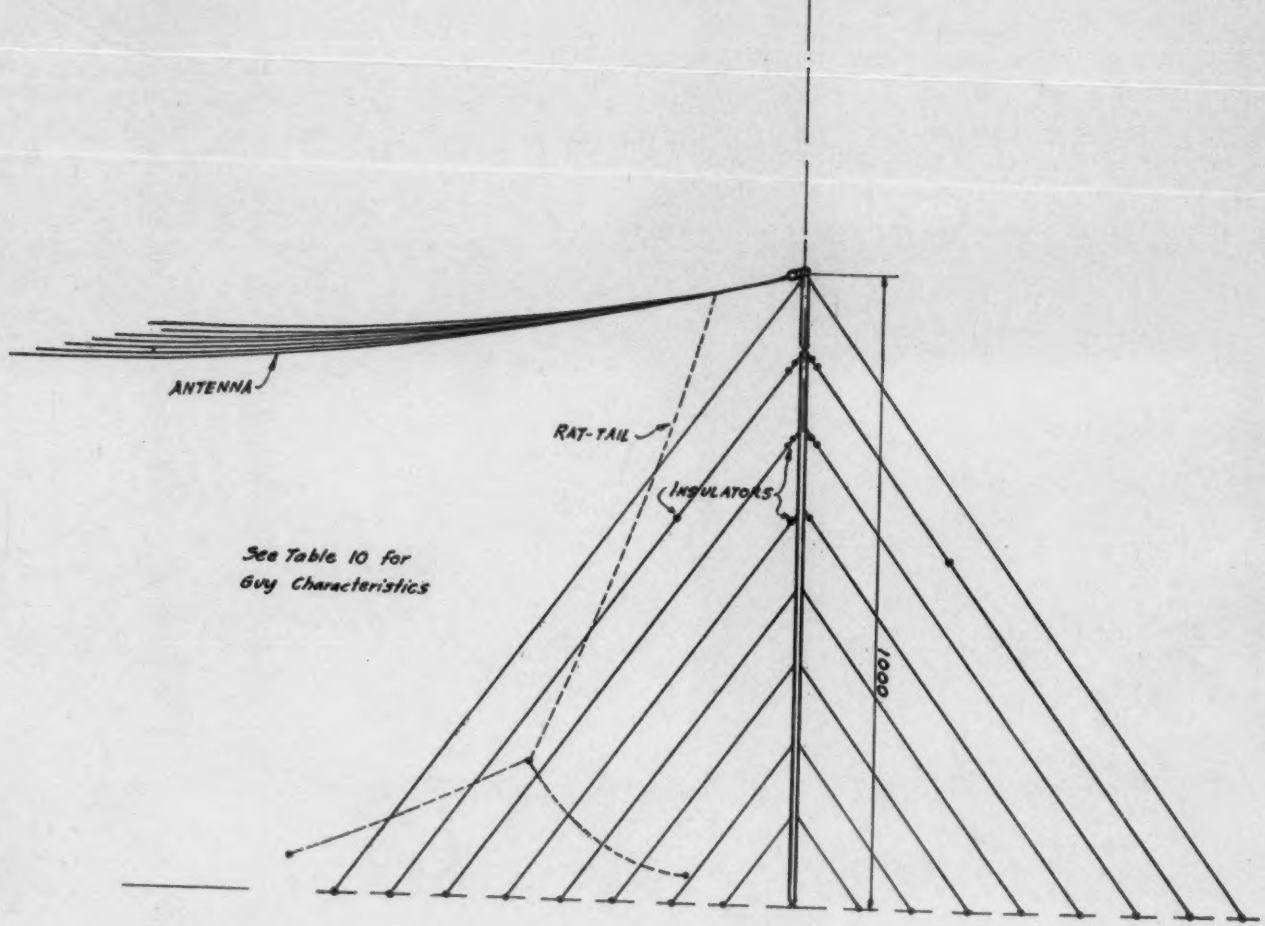


PLAN OF MAST AND ¼ ANTENNA

(NO WIND CONDITION)



ELEVATION OF MAST AND ANTENNA

FIG. 5.

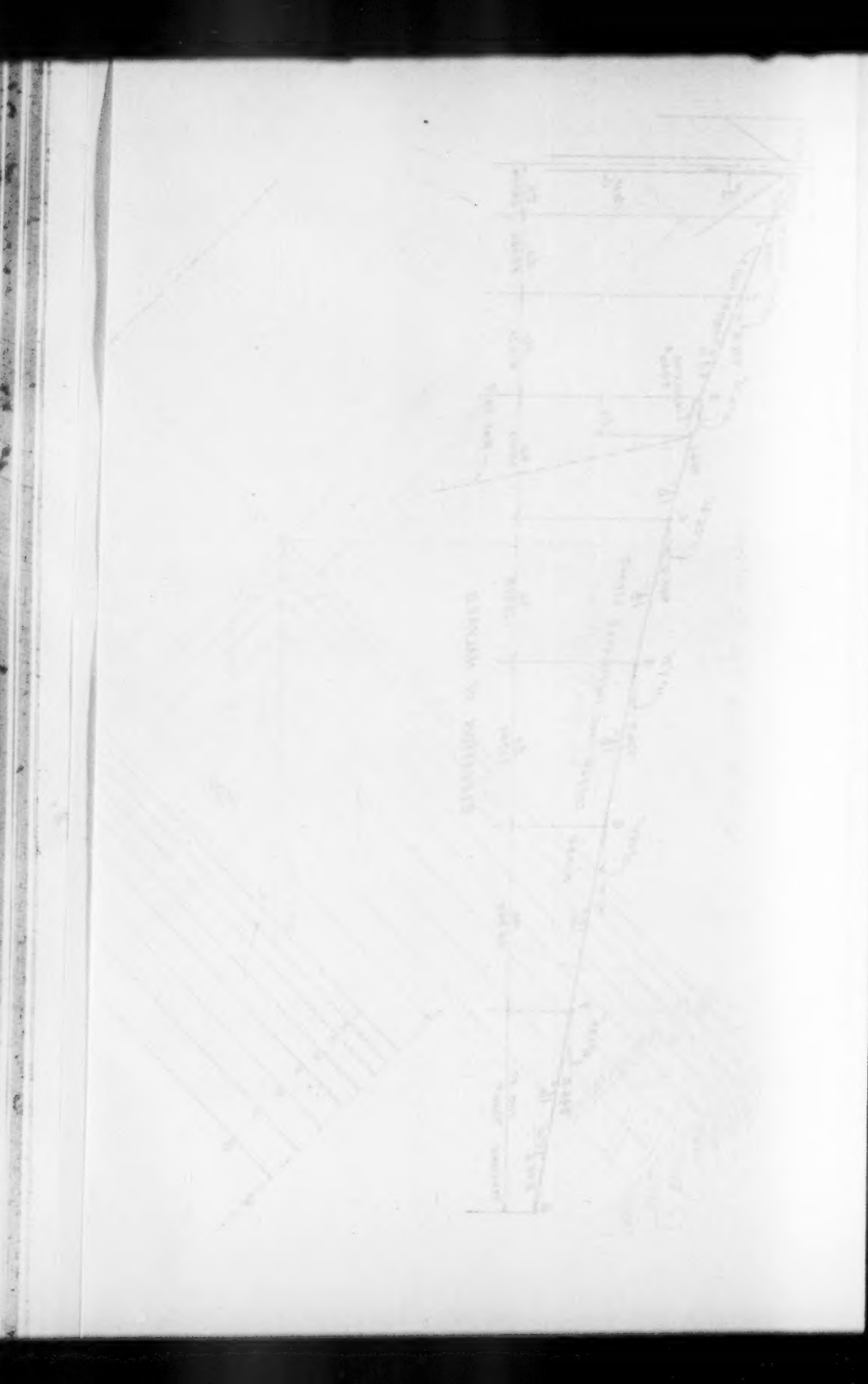


TABLE 6.
Antenna Characteristics.

Section	l	L'	L	f	n	β	W (lbs.)
8	2071.670	2075.094	2070.852	51.5	.02490	5° 41′ 12″	996.1
7	2213.092	2217.271	2212.737	58.9	.02661	6° 4′ 35″	1064.3
6	2340.372	2345.315	2340.519	65.9	.02814	6° 25′ 22″	1125.8
5	2453.510	2459.205	2454.175	72.4	.02950	6° 43′ 50″	1180.4
4	2552.510	2558.923	2553.688	78.3	.03069	6° 59′ 58″	1228.3
3a	2595.364	2602.105	2596.781	81.0	.03121	7° 6′ 58″	1248.0
3i	5.94			0.84			170.0
3e	15.06			2.42			8.0
Total 3	2637.364			84.26		9° 13′ 50″	1604.0
2a	2560.076	2566.546	2561.295	78.9	.03078	7° 1′ 10″	1232.0
2i	5.94			0.83			170.0
2e	68.06			10.92			33.0
Total 2	2708.076			90.7		9° 18′ 0″	1638.0
1a	2532.644	2541.079	2536.089	89.5	.03534	8° 2′ 42″	2120.0
1i	5.93			0.91			170.0
1e	110.07			18.90			103.2
Total 1	2764.644			109.3		10° 5′ 40″	2666.4

Notation: l —horizontal span
L'—length of curve
L —unstressed length
f —sag
n —ratio f/l
β —slope of tangent at support
W—weight

Notes: Center tension in No. 1: 7500 lbs.
Center tension in 2–8 incl.: 5000 lbs.
Insulators 1i, 2i and 3i:
two tube 7″ ϕ × 4′–6″ long
Wire 1: $\frac{5}{8}$″ ϕ extra high tensile
Copperweld stranded, 7 No. 4;
0.836 lbs. per lin. ft.
Wires 2 to 8 incl.:
$\frac{15}{32}$″ ϕ extra high tensile Copperweld
stranded, 7—0.155″;
0.481 lbs. per lin. ft.

Under dead load only, the characteristics of the antenna wires are determined from the formula:

$$H = \frac{W}{8n}, \tag{11}$$

where H is the horizontal tension at mid-span,

W the total weight,

n the sag ratio $= \frac{f}{l}$,

f the sag at mid-span,

l the horizontal span or chord length.

$$\text{Length of the curve is } L' = l\left(1 + \frac{8}{3}n^2\right). \tag{12}$$

The true or unstressed length of the wire, L, is obtained by subtracting the elastic elongation from the length of the curve.

$$\Delta L' = \frac{T_x \Delta L}{AE}, \tag{13}$$

$$T_x = H\sqrt{1 + 64\frac{f^2x^2}{l^4}}, \tag{14}$$

integrating over the entire length of the wire, after substituting the value of T_x in formula (13),

$$L' = \frac{Hl}{AE}\left(1 + \frac{16}{3}n^2\right). \tag{15}$$

The unstressed length of the wire is then

$$L = L' - \Delta L' = l\left(1 + \frac{8}{3}n^2\right) - \frac{Hl}{AE}\left(1 + \frac{16}{3}n^2\right). \tag{16}$$

From equations above, all the characteristics of the wires listed in Table 6 are determined.

The halyard consists of a series of connecting cables between the ends of the antenna wires and also includes several insulator units and the weight of the rat-tail. Vector summation of the forces acting at the lowest section of the halyard gives the amount and direction of the resultant. By assuming a value of the size of rope of known weight for the halyard section, the resultant force and slope are determined at the upper end of the section. The rope takes the shape of a parabolic curve (very approximately) and the characteristics are found by the same equations derived for sloping guy cables. The maximum stress is used for checking the assumed size of the section. The average slope, its unstressed length and the lengths of the chord and curve are determined. Similar procedure for each successive section, taking into account the insulators and rat-tail, gives the maximum halyard stress at the tower, 66,000 lbs. This determines the weight of the counterweight and also becomes the maximum stress in the halyard under all cases of wind loading. Vertical and horizontal components under wind load depend upon the geometric shape of the halyard.

The design of the antenna as a whole and in its component parts under wind load is a statically indeterminate problem and a method of approximation and check must be resorted to for a solution. The governing conditions are:

(1) The tension at the upper end of the counterbalanced halyards is constant.

(2) The true or unstressed length of each wire is constant. The wind load is assumed as a horizontal force and the magnitude is com-

puted for each wire in accordance with the formulas described in Part A, "Wind Pressure."

The vertical weights remain unchanged and as a first approximation it is assumed that the vertical components of all reactions remain the same as tabulated under dead load alone. The shifting of the antenna center of gravity is relatively a short distance and so this assumption is permissible.

Assuming a uniform distribution of load, condition (1) is then used to obtain limiting values for the reactions, combining the effect of dead load with the assumed wind pressure, in magnitude and direction, at each halyard. An assumption is then made, the choice of the true reaction, and the total load is distributed downward along the halyard and into the individual wires. Condition (2) is then used to check the assumptions.

The procedure adopted in determining stresses under wind loading was based on the following argument. The unstretched length of a wire is given in formula (16). Calling W the resultant (diagonal) of dead weight and horizontal wind loads, and the other symbols as before with the subscript denoting wind condition, the unstressed length in terms of cable characteristics under the wind load is

$$L = l_1\left(1 + \frac{8}{3}n_1^2\right) - \frac{H_1 l_1}{AE}\left(1 + \frac{16}{3}n_1^2\right). \qquad (17)$$

Under wind load, the span, sag and tension change, since the supports are not fixed. Equating the two values of the unstressed length, which must be equal, gives an exact solution for the relation between span, sag and tension under any two load conditions. The resulting expression is too complicated for application. By inspecting the results of the dead load computations, it will be noticed that an assumption, $L = l$, i.e. the chord length and unstressed length are equal under dead load, cannot bring in much error, since the values differ by only one or two feet. With this assumption,

$$L = l = l_1\left(1 + \frac{8}{3}n^2\right) - \frac{H_1 l_1}{AE}\left(1 + \frac{16}{3}n_1^2\right)$$

$$= l_1\left(1 + \frac{8}{3}n_1^2 - \frac{H_1}{AE} - \frac{16}{3}\frac{W_1^2}{64H_1^2AE}\right). \qquad (18)$$

Substituting for $n_1 = \frac{W_1}{8H_1}$,

$$\frac{l}{l_1} = 1 + \frac{8}{3}\frac{W_1^2}{64H_1^2} - \frac{H_1}{AE} - \frac{16}{3}\frac{W_1^2}{64H_1^2AE}$$

$$= 1 + \frac{W_1^2}{24H_1^2} - \frac{H_1}{AE} - \frac{W_1^2}{12H_1^2AE}. \qquad (19)$$

The last two terms are so small that they may be disregarded, especially since the wind pressure is certainly not known within the same percentage of exactness. The reduced formula becomes

$$l_1 = \frac{l}{1 + \frac{W_1^2}{24H_1^2}}, \qquad (20)$$

a relation between l_1 and H_1 for any load case. Tables were computed for each case of wind load considered, for each wire of the antenna, from which l_1 or H_1 could readily be found if the other value were known.

The problem was solved by drawing the antenna system to a large scale, assuming a displacement of a definite amount, to approximate the increase in sag of the individual wires, and drawing the new positions of the chords of the individual wires. New values of l_1 were obtained by scaling and the corresponding values of H_1 from the tables previously prepared. Summating the wind loads and the horizontal components of the tensions in the wires at the halyard intersections, vertically, resulted in a new value for the tension at the upper end of the halyard. At the counterbalanced towers, the value is equal to the counterweight. About three trials were carried to completion and plotted graphically. The amount of antenna movement or "drift" which corresponded to the proper halyard reaction was picked from the graph, adjusting for the movement of the top of the masts under the corresponding wind load.

In this way, a new location of the antenna was found which satisfied the following conditions:

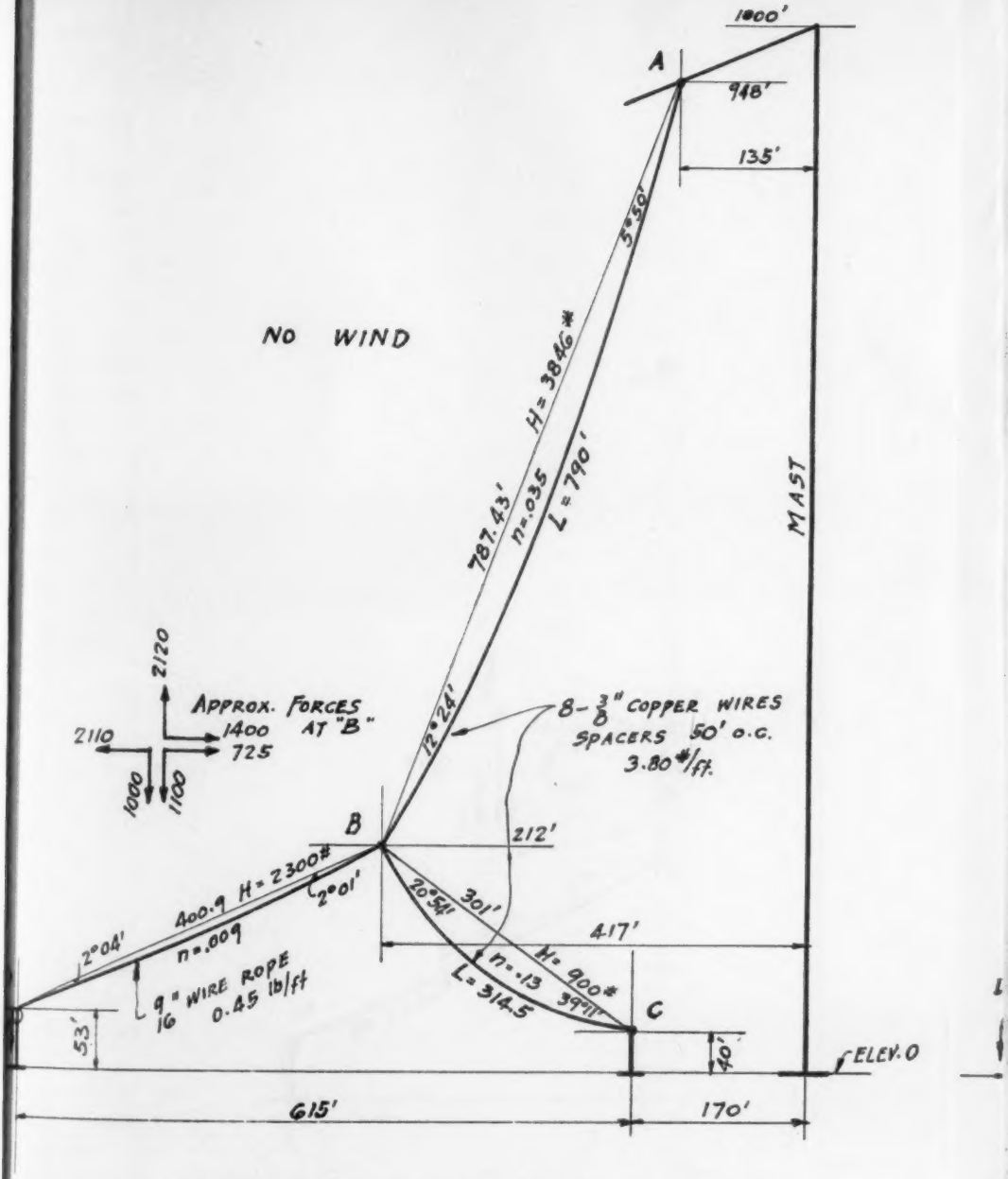
(*a*) Static—equilibrium of forces.
(*b*) Geometric—closing of the force polygons formed by the wires and halyards.
(*c*) Weight—unstressed lengths of wires remain constant.

The maximum sags were then computed. The plane of the wire is the same as the direction of the resultant loads. The components of the sag vertically and horizontally were computed. The amount of movement of the counterweight was determined from the new location of the halyards with reference to the masts.

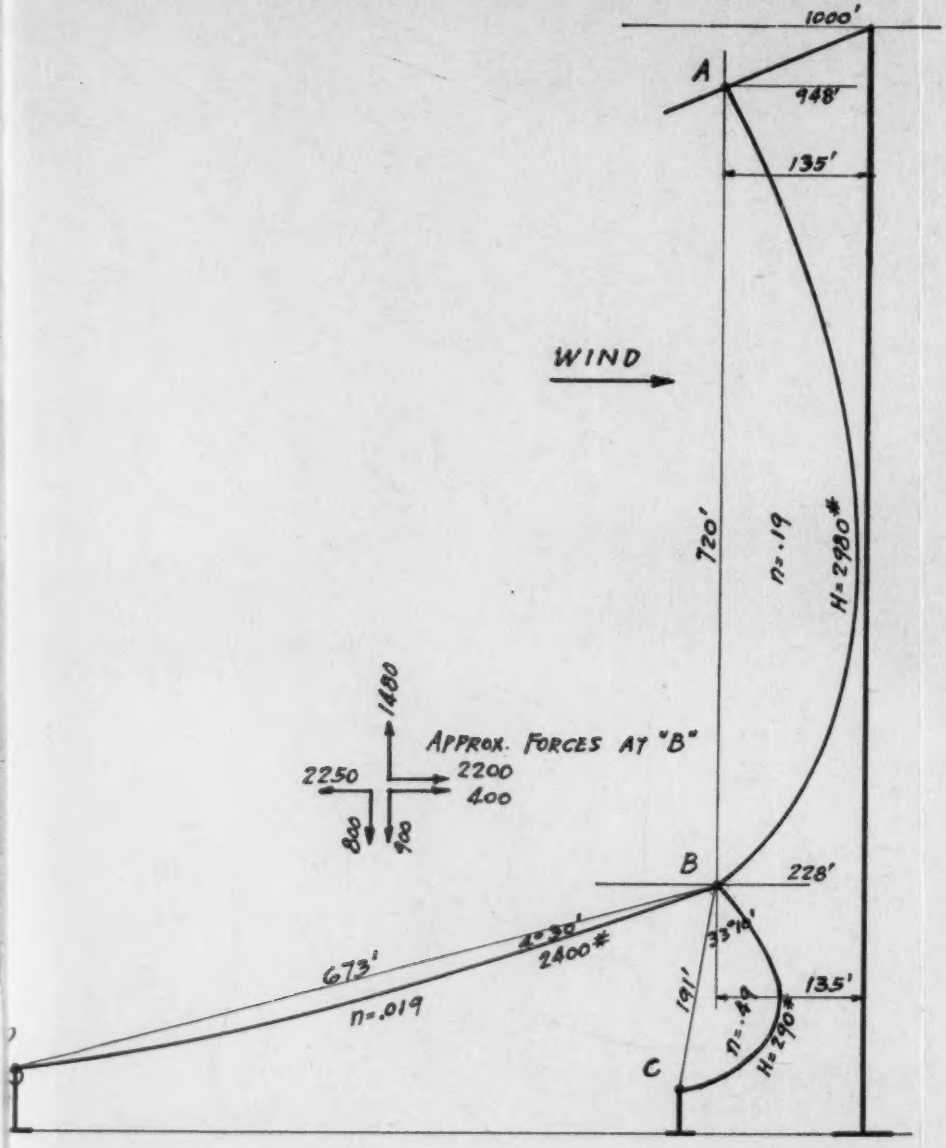
The wind conditions were assumed:

(1) Wind direction parallel to the sides of the squares.
(2) Wind direction parallel to the diagonal passing through two counterbalanced towers.
(3) Wind direction parallel to the diagonal passing through one counterbalanced and one fixed tower.

Wind pressure on antenna and halyard wires was taken at 44 lbs. per sq. ft., equivalent to a wind velocity of about 120 m.p.h., and was



RAT- TAIL UNDER DEAD LOAD



WIRE	AVE. WIND VELOCITY	PRESSURE
AB	105.5 MPH	30.8 lb/sq ft.
BC	89.5	22.2
BD	89	22.0

APPROXIMATE SHAPE AND FO

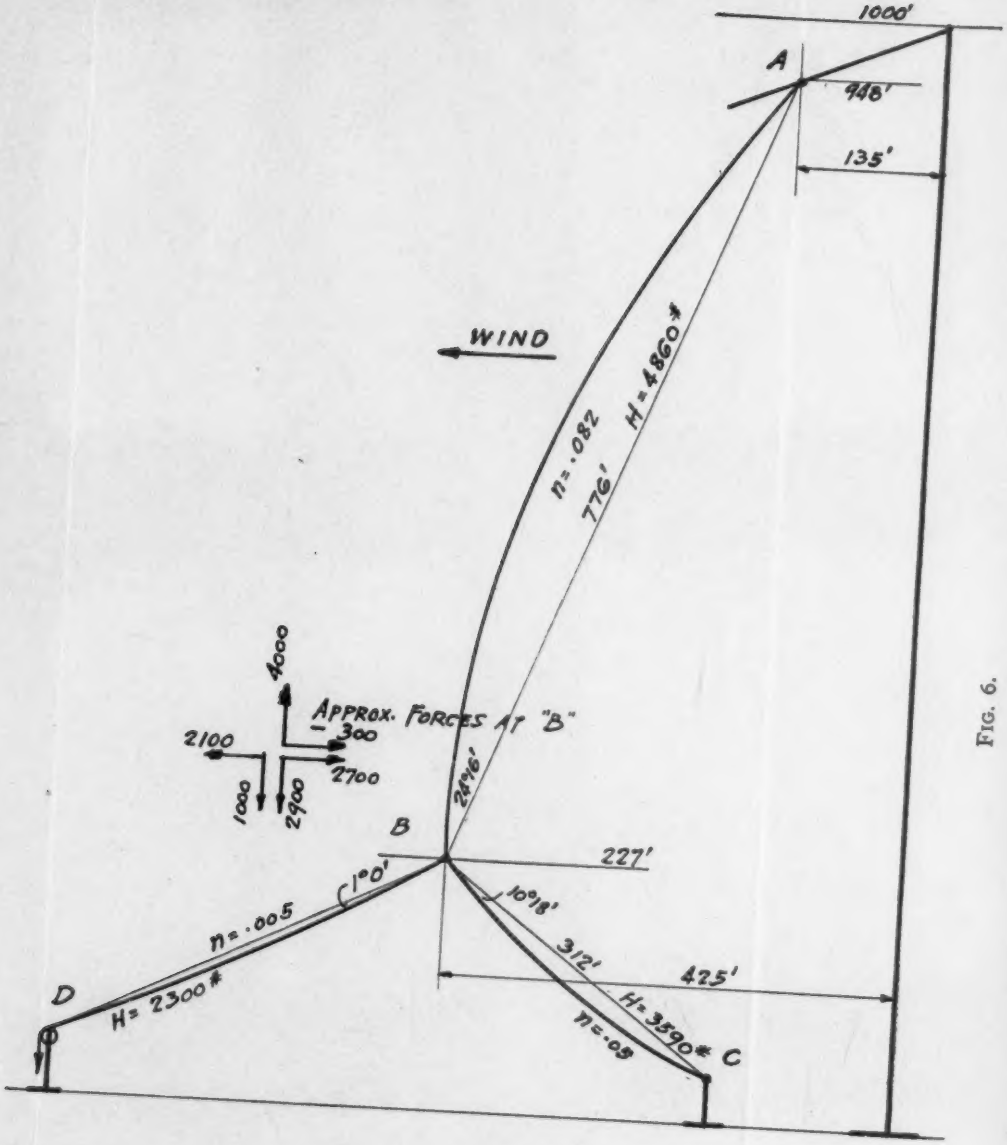
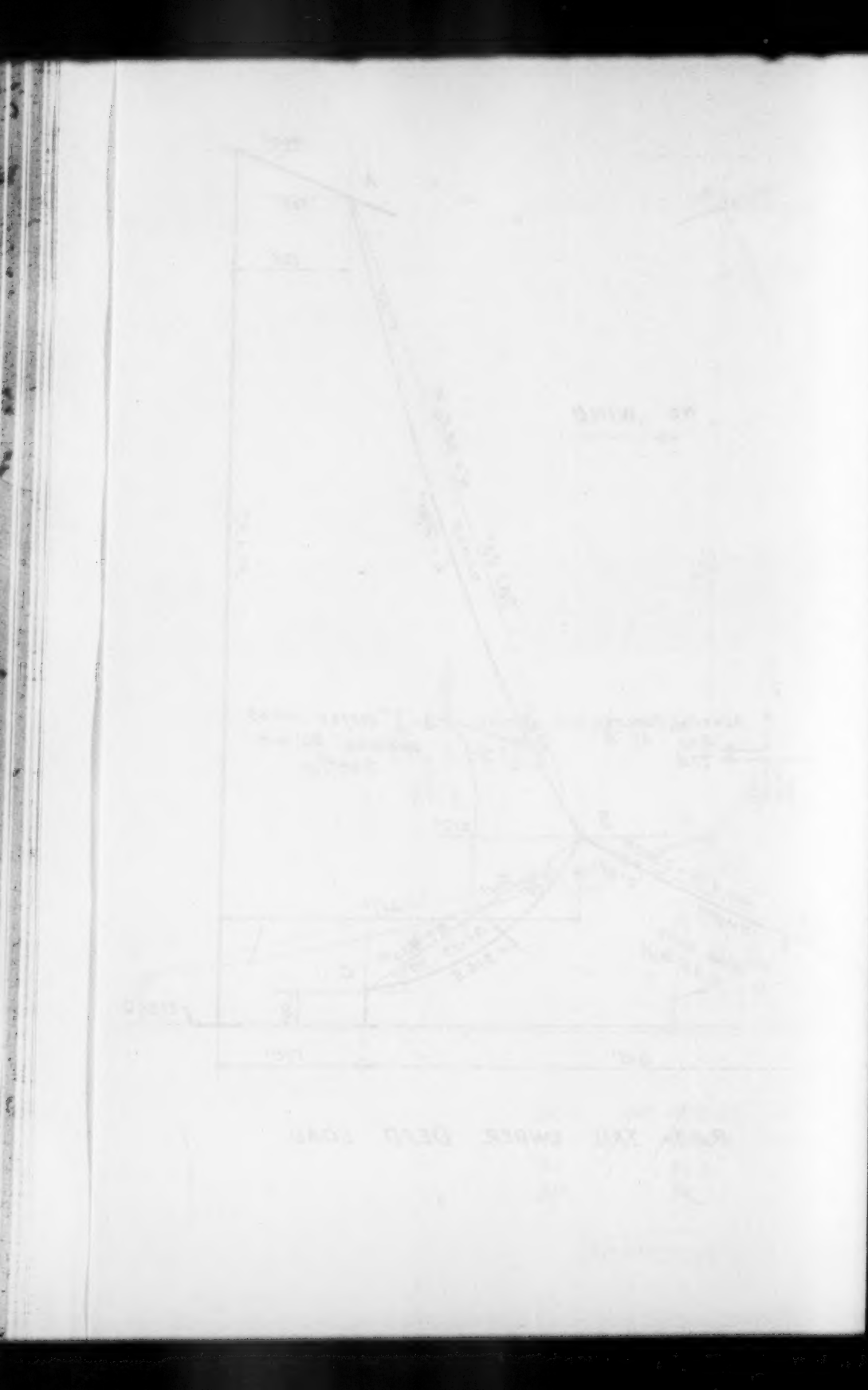


FIG. 6.

ESSURE	NORMAL LOADS ON CHORDS
8 lb/sq ft.	6.26 lb/ft.
.2	5.00
.0	0.50

AND FORCES UNDER WIND LOADS.



assumed the same for all conditions and as an average for the elevation range including the antenna system. This pressure was reduced by $\sin^2 \phi$ for the wires oblique to the wind direction. The same wind pressure was taken on each insulator over the projected exposure normal to the wind direction. The area of the exposure was determined by scaling carefully drawn diagrams of the irregularly shaped insulators.

Since wind exposure and weight of wires are constant and since wires No. 8 to No. 2 are the same, the problem was considerably simpler than the description of the method may indicate.

For wind normal to a wire span, for No. 8 to No. 2 incl., the ratio of vertical to horizontal load is 0.481 to 1.706. Since the $\sin^2 45° = \frac{1}{2}$, the ratio for wind at 45° to a wire span is 0.481 to 0.853. The halyard sections are all different and each section required separate computation. Wind direction coinciding with a wire span increased the sag, made the curve unsymmetrical but in a vertical plane. The chord lengths and tensions in the antenna wires under the first wind loading, 44 lbs. per sq. ft. acting parallel to the side of the square, are tabulated in Table 7.

TABLE 7.

Antenna Chord Lengths and Tensions at Mid Spans Under 44 lbs. per sq. ft. Wind Load Parallel to the Side of the Square.

Wire No.	Leeward Set		Windward Set		Each Side Set	
	l_1	H_1	l_2	H_2	l_3	H_3
8	2012	4200	2046	6400	2062	2925
7	2150	4740	2187	7350	2204	3330
6	2260	4580	2308	7100	2327	3090
5	2360	4530	2415	7030	2438	2940
4	2455	4700	2512	7300	2519	2155
3	2533	4830	2597	7500	2604	2300
2	2602	4920	2653	7050	2676	2445
1	2649	7500	2728	11700	2705	3200

The distorted shape of the antenna under wind load (with towers out of plumb) tensions is seen from the tabulated values. Maximum tensions occur at the ends of the spans, are computed for design of wire sizes.

The "rat-tail" hangs from the halyard to the ground, and wind pressures were taken at average unit values for each of the three portions. Fig. 6 shows the shape and the cable characteristics under dead load alone and under two wind directions. The lower end of the guy wire is connected to a small counterweight, 2,300 lbs. weight moving vertically in the tower. The lower end of the conductor cable is fixed.

It should be noted that the physical properties of the wires and cables used must be known. The problem is similar to the design of single transmission lines. The tables prepared by James S. Martin (23) for

TABLE 8.

Detailed Calculation of Stresses in Guy Cables.

Line	Item	Dimension	Symbol	Formula	Value for VII	Value for VIII
1	Cable diam.	in.	d	—	$1\frac{1}{4}''$	$1\frac{1}{4}''$
2	Grade of steel	—	—	—	Plough	Steel
3	Weight	lb./ft.	r	—	3.901	5.497
4	Breaking strength	M lb.	S	—	192.0	271.3
5	Elastic modulus × area	lb.	EA	—	24×10^6	34×10^6
6	Inclination to horizontal		ϕ		45°	45°
7	Elevation of guy point	ft.	h		883.25	991.87
8	Chord length (no wind)	ft.	l_0	$h\sqrt{2}$	1,249.3	1,402.5
*9	Normal component of weight	lb.	W_0	$rl_0/\sqrt{2} = rh + I$	3.762	5.930
10	Horizontal mast movement of guy point,					
	in direction of wind	ft.	u_x		5.37	6.75
11	in direction normal to wind	ft.	u_y		0	1.40
12	Wind pressure on mast	lb./sq. ft.	w_m		66.1	71.7
13	Ave. wind on guy	lb./sq. ft.	w_g		43	46
14	Mast area exposure	sq. ft./ft.	A		4.9	4.9
*15	Total wind on mast at guy	lb.	W_m	$\frac{1}{2}Aw_m(h_{n+1} - h_{n-1})$	32,950	21,980
*16	Normal component of wind,					
	on guys in wind direction	lb.	W''	$0.94\, w_g l_0$	6,660	8,500
17	on side guys	lb.	W'''	$w_g l_0$	7,060	8,680
18	Total normal load on					
	windward guy (1)	lb.	W_1	$W'' + W_0$	10,422	14,430
19	leeward guy (2)	lb.	W_2	$W'' - W_0$	−2,898	−2,570
20	each side guy (3) (4)	lb.	W_3	$\sqrt{W_0^2 + W'''^2}$	8,020	10,500
*21	Antenna pull (guy VIII only)					
22	parallel to wind	lb.			—	51,000
23	normal to wind	lb.			—	17,000
24	Initial guy tension—minimum	lb.	H_0		19,750	31,350
25	Initial sag ratio		n_0	$f_0/l_0 = W_0/8H_0$	0.0228	0.0228
26	Slope angle of guy—no wind		β_0	arc tan $4n_0$	5° 13′	5° 13′
*27	Elongation guy (1)— force parallel to wind	ft.	Δl_{1x}	$\frac{au_x}{l_0} - \frac{a^2u_x^2}{2l_0^3}$	3.767	4.750
28	Elongation guy (1)— normal to wind	ft.	Δl_{1y}	$u_y^2/2l_0$	0	0.0034
29	Total elongation guy (1)	ft.	Δl_1	$\sqrt{\Delta l_{1x}^2 + \Delta l_{1y}^2}$	3.767	4.750
30	Elongation guy (2)— force parallel to wind	ft.	Δl_{2x}	$-\frac{au_x}{l_0} - \frac{a^2u_x^2}{2l_0^3}$	−3.828	−4.770
31	Elongation guy (2)— normal to wind	ft.	Δl_{2y}	$-u_y^2/2l_0$	0	0.0035
32	Total elongation guy (2)	ft.	Δl_2	$\sqrt{\Delta l_{2x}^2 + \Delta l_{2y}^2}$	−3.828	−4.770
33	Elongation guy (3)— force parallel to wind	ft.	Δl_{3x}	$u_x^2/2l_0$	0.00215	0.017
34	Elongation guy (3)— normal to wind	ft.	Δl_{3y}	$\frac{au_y}{l_0} - \frac{a^2u_y^2}{2l_0^3}$	0	0.985
35	Total elongation guy (3)	ft.	Δl_3	$\sqrt{\Delta l_{3x}^2 + \Delta l_{3y}^2}$	0.00215	0.990
36	Elongation guy (4)— force parallel to wind	ft.	Δl_{4x}	$u_x^2/2l_0$	0.00215	0.017
37	Elongation guy (4)— force normal to wind	ft.	Δl_{4y}	$-\frac{au_y}{l_0} - \frac{a^2u_y^2}{2l_0^3}$	0	−0.985
38	Total elongation guy (4)	ft.	Δl_4	$\sqrt{\Delta l_{4x}^2 + \Delta l_{4y}^2}$	0.00215	−0.990

* Notes on lines:

9 Weight of guys plus insulators.

15 h_{n+1} is elevation of next higher guy; h_{n-1}, of next lower.

16 Factor $0.94 = \frac{2 \sin \beta}{1 + \sin^2 \beta}$ for 45°, the Duchemin formula;

see text for comparison of results with use of $\sin^2 \phi$ factor.

21 Valve of antenna pull used in the computation in this example.

27 a = Horizontal projection of guy = h for $\phi = 45°$.

TABLE 8.—*Continued.*

Line	Item	Dimension	Symbol	Formula	Value for VII	Value for VIII
39	Algebraic expressions		N_0	$\frac{H_0}{EA} - \frac{8}{3} n_0^2$	−0.00057	−0.000463
40			ΔN_1	$\Delta l_1/l_0$	0.00301	0.00342
41			ΔN_2	$\Delta l_2/l_0$	−0.00306	−0.00343
42			ΔN_3	$\Delta l_3/l_0$	0.0000017	0.00071
43			ΔN_4	$\Delta l_4/l_0$	−0.0000017	−0.00071
44			N_1	$N_0 + \Delta N_1$	0.00244	0.00296
45			N_2	$N_0 + \Delta N_2$	−0.00363	−0.00380
46			N_3	$N_0 + \Delta N_3$	−0.00057	0.00025
47			N_4	$N_0 + \Delta N_4$	−0.00057	−0.00117
48	Sag ratio—guy (1)	n_1	from	$n_1^2 + \frac{8}{3} N_1 n_1 = \frac{3W_1}{64EA}$	0.0171	0.01532
49	guy (2)	n_2	from	$n_2^2 + \frac{8}{3} N_2 n_2 = \frac{3W_2}{64EA}$	0.0388	0.0394
50	guy (3)	n_3	from	$n_3^2 + \frac{8}{3} N_3 n_3 = \frac{3W_3}{64EA}$	0.0285	0.0234
51	guy (4)	n_4	from	$n_4^2 + \frac{8}{3} N_4 n_4 = \frac{3W_4}{64EA}$	0.0285	0.0304
52	Sag—no wind	ft.	f_0	$n_0 l_0$	28.3	31.3
53	guy (1)	ft.	f_1	$n_1 l_1$	21.4	20.8
54	guy (2)	ft.	f_2	$n_2 l_2$	48.4	55.0
55	guy (3) parallel to wind	ft.	f_3	$n_3 l_3$	35.6	32.5
56	guy (3) total	ft.	c_3		45.5	45.1
57	guy (4) parallel to wind	ft.	f_4	$n_4 l_4$	35.6	42.2
58	guy (4) total	ft.	c_4		45.5	52.6
59	Slope angle at end (1)		β_1	arc tan $4n_1$	3° 55′	3° 30′
60	Slope angle at end (2)		β_2	arc tan $4n_2$	8° 40′	8° 57′
61	Slope angle at end—parallel to wind (3)		β_3	arc tan $4n_3$	6° 30′	5° 21′
62	Slope angle at end—parallel to wind (4)		β_4	arc tan $4n_4$	6° 30′	6° 56′
63	Slope angle at end—total (3)		δ_3	arc tan $\frac{\sqrt{f_0^2 + f_3^2}}{\frac{1}{2} l_0}$	4° 10′	3° 40′
64	Slope angle at end—total (4)		δ_4	arc tan for f_4	4° 10′	4° 20′
65	Max. tension—no wind	lb.	T_0	$H_0 \sec \beta_0 + \frac{1}{2} W_0 \tan \phi$	21,710	33,500
66	Min. tension—guy (1)	lb.	H_1	$W_1/8n_1$	77,000	120,900
67	Max. tension—guy (1)	lb.	T_1	$H_1 \sec \beta_1 + \frac{1}{2} W_0 \tan \phi$	79,060	123,200
68	Factor of safety—guy (1)		F	S/T_1	2.43	2.20
69	Min. tension—guy (2)	lb.	H_2	$W_2/8n_2$	9,500	7,800
70	Max. tension—guy (2)	lb.	T_2	$H_2 \sec \beta_2 + \frac{1}{2} W_0 \tan \phi$	11,500	10,000
71	Min. tension—guy (3)	lb.	H_3		36,240	57,600
72	Max. tension—guy (3)	lb.	T_3		38,330	59,950
73	Min. tension—guy (4)	lb.	H_4		36,240	43,300
74	Max. tension—guy (4)	lb.	T_4		38,330	46,450
75	Horizontal force parallel to wind	lb.		on antenna	—	51,000
76		lb.		on mast section	32,950	21,980
77		lb.		on guys	8,300	10,420
78		lb.		on guy insulators	930	1,330
79	Total horiz. force parallel to wind			Lines 75+76+77+78	42,180	84,730
80	Horiz. force normal to wind	lb.		on antenna	—	17,500
81	Sum of horiz. components T_1 & T_2	lb.			42,700	84,400
82	Vertical component of T_1	lb.	V_1		52,000	92,500
83	Sum of vertical comps. of all Ts	lb.	V		108,800	181,000

sag, length and stretch are useful tools in reducing the laborious computations.

Loads on Masts.

The loads acting on the guyed masts under no wind conditions are:

1. Dead weight of steel work.
2. Vertical reactions of the guys; horizontal reactions balance at each point of connection.
3. Weight of counterweight, sheaves and head frame.

4. Horizontal and vertical components of halyard reaction, included in reactions of top set of guys.

Under wind load conditions, the only forces which change are the vertical reactions of the guys. In addition there are also:

5. Lateral forces on the mast structure causing cross bending.
6. Eccentricity from unequal guy reactions at each level.
7. Twist at the top from the unsymmetrical reaction of the halyard.

Since usual structural design methods were used, only some of the major unusual features will be described.

Antenna Loads on Masts.

To determine maximum reactions from the antenna on the mast, a tabulation was made for an exterior mast and an interior mast, for 120 m.p.h. and 150 m.p.h. winds from three assumed directions. The total horizontal pull is always 62,000 lbs. from each antenna halyard which is counterweighted. The direction of the wind affects the resultant direction of the halyard and, correspondingly, the components to be balanced by the guys at the top of the mast. The total wind load in the antenna acting at each mast is 33,730 lbs. under 150 m.p.h. and 21,600 lbs. under 120 m.p.h. velocities. The direction of the wind affects the wind reactions to be balanced by the guys at the top of the mast. Table 9 shows the halyard and wind reactions in each pair of mast guys. It is tabulated on an assumed compass orientation for convenience only.

The maximum horizontal load to be taken by a pair of opposite guys is 69,100 lbs. under 120 m.p.h. wind and 100,600 lbs. under 150 m.p.h. wind. The latter case is not considered, since the condition of 150 m.p.h. wind over the entire antenna exposure is not probable.

D. DESIGN OF GUYS FOR TOWERS.

Under no wind, the design is for a vertical mast with 8 sets of guys at fixed locations, all chords and stresses in each set of four being identical. Under wind loads, the symmetrical state changes and guys become unequal in chord length from the lateral movement of panel points. A requirement that under maximum wind the mast must remain substantially straight with a maximum deviation of 7.5 feet at the top, was the controlling factor in the design of the guys. A detailed study of the mast shape under wind load was made under the following:

(1) Rocker pivots in the mast at each guy panel point.
(2) Pivots at the base and at panel points VI and VII.
(3) Pivots at the base and at panel point VII.
(4) Pivots at the base and at panel point VI.
(5) Pivot at base only.

The conclusion was reached to provide pivots at panel points 0 and VII only, because the last case showed extremely high bending stresses in the mast and the other cases indicated unsatisfactory distortions. To reduce the laborious work of determining stresses in the guys for the great number of trial designs under various wind loads, a systematic

TABLE 9.

Horizontal Antenna and Wind Reactions at Top of Masts.

Guys are located in North, East, South and West quarters. Halyards under no wind are located at S.45°W. and for the interior mast, also at N.45°E.

Case	Wind Velocity	Assumed from	Halyard Position	Horizontal Forces (in 1000 lb. units)					
				N–S Direction			E–W Direction		
				Halyard	Wind	Total	Halyard	Wind	Total
A1	0	—	S45W	43.8	—	43.8	43.8	—	43.8
B1	120	N	S40W	47.5	21.6	69.1	39.8	—	39.8
C1	120	E	S50W	39.8	—	39.8	47.5	21.6	69.1
D1	120	S	S90W	—	−21.6	−21.6	62.0	—	62.0
E1	120	W	S0W	−62.0	—	−62.0	—	21.6	21.6
F1	150	N	S35W	50.8	33.7	84.5	35.6	0	35.6
G1	150	S	N75W	−33.7	−16.1	−49.8	60.0	0	60.0
A1 to G1 incl. are for an exterior mast.									
A2	0	—	S45W N45E	43.8 −43.8	—	0	43.8 −43.8	—	0
B2	120	N	S40W N90E	47.5 0	21.6	69.1	39.8 −62.0	—	−22.2
C2	120	E	S50W N0E	39.8 −62.0	—	−22.2	47.5	21.6	69.1
D2	120	S	S90W N40E	0 −47.5	−21.6	−69.1	62.0 −39.8	—	22.2
E2	120	W	S0W N50E	62.0 −39.8	—	22.2	0 −47.5	−21.6	−69.1
F2	150	N	S35W S75E	16.1 50.8	33.7	100.6	−60.0 35.6	—	−24.4
G2	150	S	N75W N35E	−16.1 −50.8	−33.7	−100.6	60.0 35.6	—	24.4
A2 to G2 incl. are for an interior mast with 2-halyards.									

tabular arrangement, shown in Table 8, was developed. As an example of the method, the final values for the guys VII and VIII are listed, together with the successive formulas.

Table 10 lists the eight sets of guys with dead load and wind load stresses. Concrete anchors were provided to resist safely the maximum tensions in each of the 32 guy cables.

The values in Tables 9 and 10 are based on the wind pressure formula:

$$p = 0.0033V^2 \quad \text{and} \quad p_n = \frac{p^2 \sin \phi}{1 + \sin^2 \phi}. \tag{9a}$$

A study was made for guy VII to determine the effect on the stress computations by changing the wind pressure formulas to

$$p = 0.00277V^2 \quad \text{and} \quad p_n = p \sin^2 \phi, \tag{10a}$$

which latter seem to be in better agreement with recent tests (see Tables 4 and 5). It was found that the change in pressure conversion formulas increased the factor of safety by 20 per cent. and reduced the indicated total vertical load on the mast by 12 per cent. The

TABLE 10.

Characteristics and Stresses in Mast Guys.

Set	Elevation of Connection to Mast	No Wind		Max.			Min.		
		H_0	n_0	T_1	H_1	n_1	T_2	H_2	n_2
VIII	1,004	31,350	.0228	123,200	120,900	.01532	10,000	7,800	.0394
VII	885	19,750	.0228	79,060	77,000	.0171	11,500	9,500	.0388
VI	783	19,110	.0165	66,300	65,200	.0133	10,050	9,000	.0258
V	675	15,550	.0176	58,100	57,350	.0123	6,500	5,400	.0297
IV	561	13,000	.0174	54,800	53,450	.0105	4,300	3,330	.0332
III	441	9,700	.0183	48,900	47,800	.0087	2,480	1,840	.0341
II	309	6,280	.0199	41,550	41,000	.0069	1,100	700	.0357
I	165	2,850	.0231	34,930	34,400	.0039	340	80	.0385

Notes:

Cables in Set VIII are $1\frac{3}{8}$" plough steel, $EA = 34 \times 10^6$.
Cables in Set VII are $1\frac{1}{8}$" plough steel, $EA = 24 \times 10^6$.
Cables in Sets VI, V, IV and III are $1\frac{1}{4}$" plough steel, $EA = 20 \times 10^6$.
Cables in Sets II and I are $1\frac{1}{4}$" crucible steel, $EA = 20 \times 10^6$.

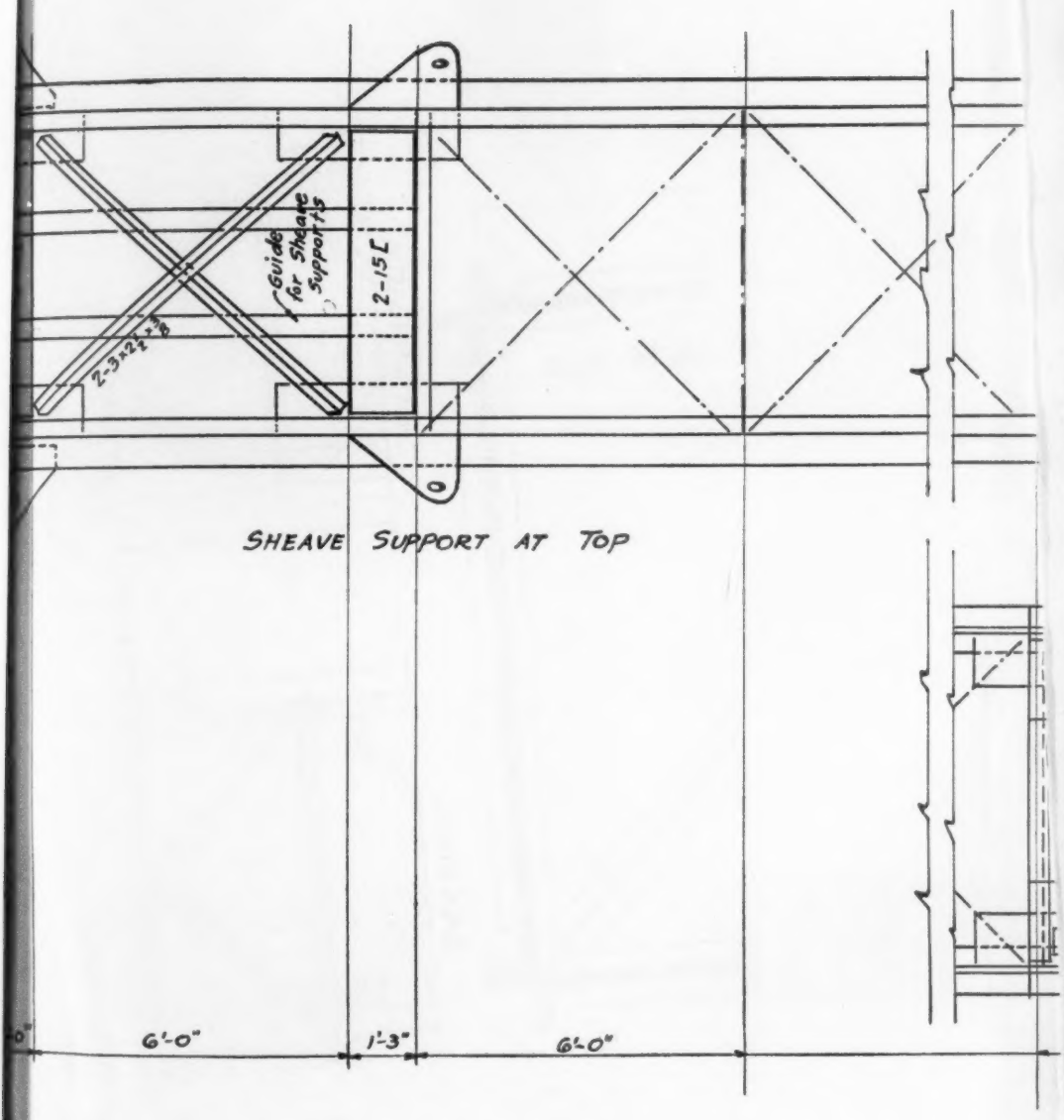
vertical component of the guy carrying maximum load under wind was reduced from 62,000 to 52,000 lbs.

The assumed wind velocity converted to load by formula (9*a*) increases guy tensions to four times dead load tension for the guys near the top of the mast where a deflection is permitted. The lowest guys, affected by the relative rigid mast, show increases to as high as 10 times dead load tension. The structural features of the guyed mast are shown in Fig. 7.

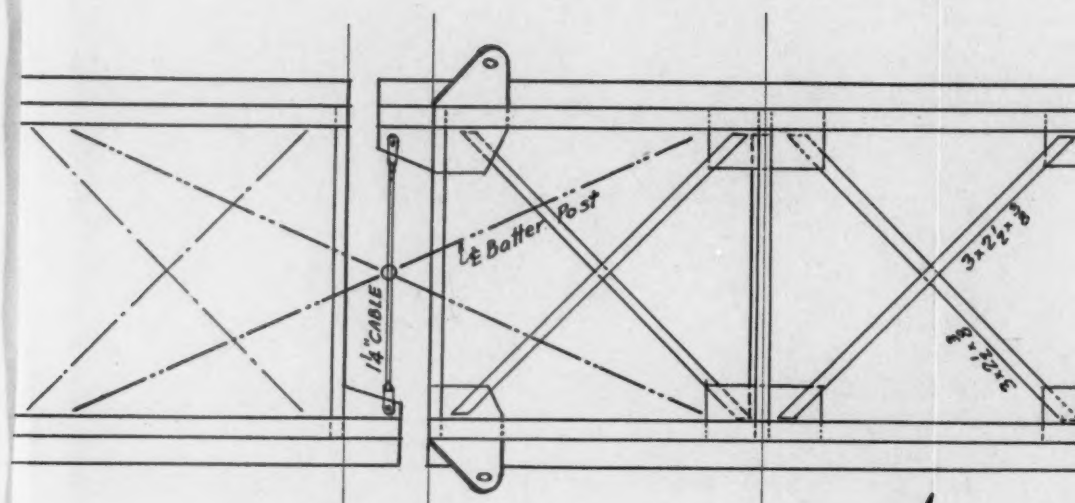
E. STRESSES IN STRUCTURAL MAST.

Under no wind, the four guys in each set have equal tensions and the total vertical force carried by the mast is symmetrically divided into the four legs. The horizontal pull of the antenna is resisted by the top guys and all guy reactions balance.

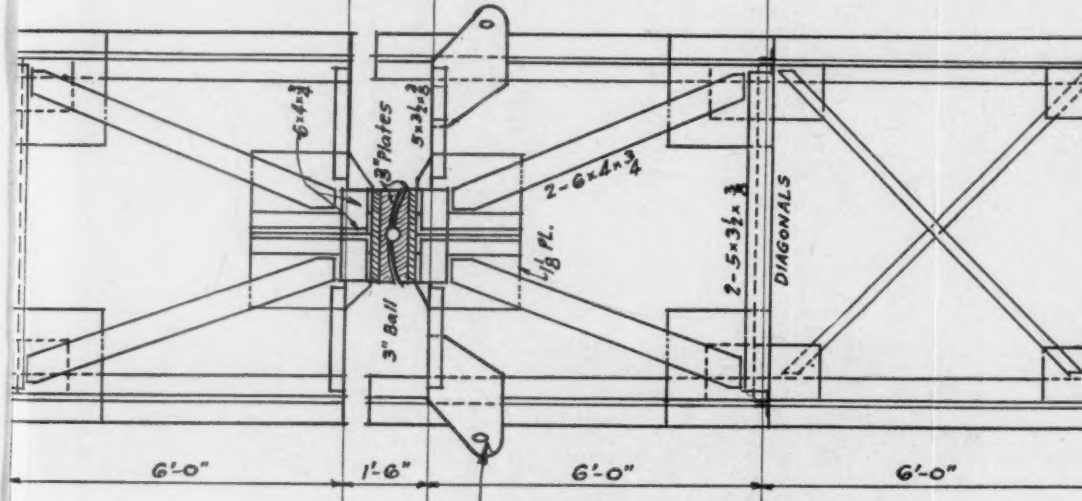
Under wind loading, the antenna reactions do not change mast



SHEAVE SUPPORT AT TOP



ROCKER & GUY POINT TYPICAL PANEL



Guy Pin Plate passes thru notch in corner angle - splice angles not shown

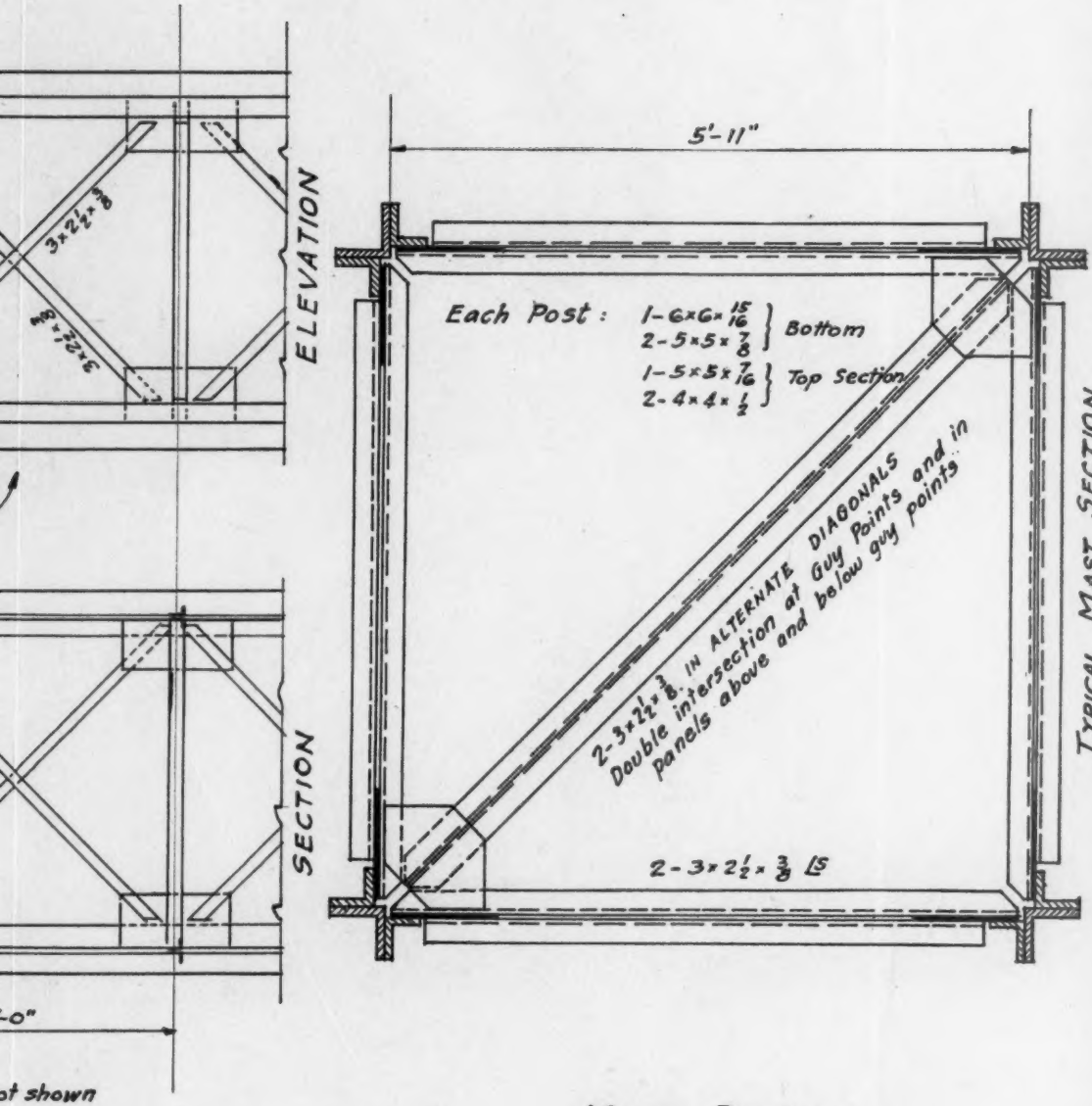


FIG. 7.

MAST DETAILS

stresses, but the unequal guy reactions at each level and the cross bending from the wind on the mast structure complicate stress determination. At the guy connection levels, the vertical component of each guy goes directly into the leg of the mast to which the guy is connected. In a section of the mast ending or resting on a rocker plate, the vertical loads in all four legs at the rocker must be equal. The stresses from wind loading are determined from normal formulas for a simple supported truss. The maximum post stress is the sum of the vertical loads at the windward guy (1) plus the wind bending.

On the assumption that under maximum wind load the mast deviates from the plumb but remains straight between the bottom and the rocker pivot at the VII set of guys, a fairly accurate determination can be made of leg stresses in the entire mast below the upper rocker. The total vertical load coming from above the rocker is equally divided into the four posts of the mast. The wind load is applied to the mast as a continuous truss with end supports simply connected and the wind stress is computed for each post, as is shown diagrammatically in Fig. 8. The weight of each post accumulates vertically. To the average vertical components of guy tensions must be added the excess of actual guy load (plus or minus) over the average at each of the four posts. Only the loads on the windward post are shown, since they represent the maximum condition and control the design. Each excess guy load varies in linear fashion from the point of connection to zero at the rocker pivot at the base of the mast. Load summations at panel points and at mid-panel points (where wind stress is maximum) indicate maximum stresses.

The inequality of loading in the four posts is equivalent to an eccentricity of the total load on the mast. The resulting moment is balanced by an adjustment of guy tensions, providing the horizontal couples to balance the moment from eccentricity of vertical load. The adjustment decreases T_1 (the windward or maximum guy tension) and increases the others, at all pivot panel points. Similar adjustments will occur if the mast does not remain straight, as will happen under any wind loads which deviate from those assumed in the computations. The maximum computed unit stress of 17,000 lb. per sq. in. under combined maximum loadings provides sufficient leeway for localized stress increases from mast deflections.

The shear stress at any section is a combination of wind pressure shear and eccentricity of the vertical loads in the four posts, the study being made for each mast section between guy panel points.

At the two pivots in the mast, the four posts are bent together into a pyramid, using four angles $6 \times 4 \times \frac{3}{4}$ for each post. The rocker plates are 2″ thick with radii of 80″ and 130″.

The mast detail is generally an assembly of six foot cubes, the vertical posts each consisting of three angles and the horizontal members of

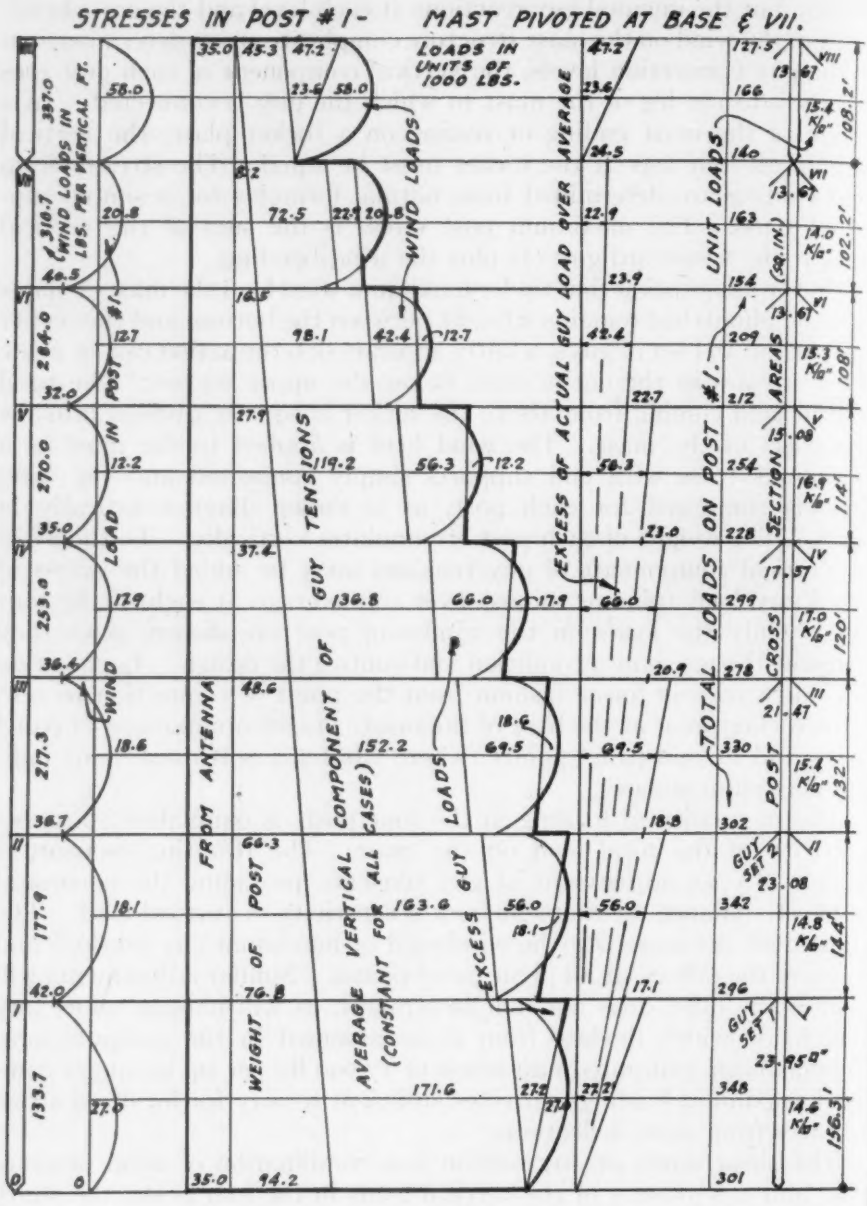

STRESSES IN POST #1 - MAST PIVOTED AT BASE & VII.
LOADS IN UNITS OF 1000 LBS.
WIND LOADS IN LBS. PER VERTICAL FT.
WIND LOAD IN POST #1
WIND LOADS
GUY TENSIONS
FROM ANTENNA
WEIGHT OF POST
AVERAGE VERTICAL COMPONENT OF GUY LOADS (CONSTANT FOR ALL CASES)
EXCESS GUY LOADS
EXCESS OF ACTUAL GUY LOAD OVER AVERAGE
TOTAL LOADS ON POST #1.
UNIT LOADS (SQ. IN.)
SECTION AREAS
CROSS POST
GUY SET 2
GUY SET 1

FIG. 8.

single angles. Single angles are placed across the diagonals of all sides of the cube.

Guys are connected to the mast by an open socket and a turned pin to a gusset connection, consisting of a flat central plate riveted to

the horizontal diagonals in the mast and two bent plates riveted to the post angles.

The foundation for a guyed mast presents no unusual problem and in this case consisted of a concrete mat supported by wood piles.

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NOTATIONS USED IN THE PAPER.

A: cross section area of cable or wire.
d: diameter of wire or width of flat plate.
E: elastic modulus.
f: sag at maximum deflection, normal to chord.
H: tension parallel to chord.

H_0: tension parallel to chord under dead load.
$H_{1,2,3}$: tension parallel to chord under combined dead and wind load.
k: coefficient in formula $p = kV^2$.
L: true or unstressed length of cable or wire.
l: span or chord length.
$l_{1,2,3}$: span or chord length under wind load.
L': length of curve.
$\Delta L'$: increase in length of curve.
n: sag ratio f/l.
n_0: sag ratio under dead load.
$n_{1,2,3}$: sag ratio under wind load.
p: horizontal wind intensity.
p_n: normal wind intensity.
P: pressure of wind.
P_g: pressure of wind at ground level.
r: radius of wire or cable.
T: maximum tension in cable or wire.
T_x: tension at intermediate point.
V: velocity of wind at elevation H.
v: velocity of wind at elevation h.
V_g: velocity of wind at ground level.
V_m: mean hourly velocity of wind.
V_t: velocity of gusts.
W: total weight of cable or wire.
$W_{1,2,3}$: resultant total load on cable under wind.
β: angle end of slope to chord.
ϕ: angle between wind and resisting surface or axis of wire.

HOW ILLUMINANTS ARE BORN.*

BY

SAMUEL G. HIBBEN,

Director of Applied Lighting, Westinghouse Lamp Division.

The generation of light, and of radiant energy closely akin to light, has throughout the past eons of man's practical experience been generally limited to the incandescence of hot solids—the minute particles of carbon in the fagot or candle flame, the white hot bits of ceria and thoria in the incandescent gas mantle, or the tungsten filament of an electric lamp bulb which glows dull red at approximately 720° K. (450° C.) and reaches its point of optimum economic light emission at about 3000° K.

All of these methods have been terribly inefficient since the intermediate step was the generation of heat, and the heat losses were on the order of 90 per cent. of the input energy! For a commercial tungsten filament (60 watt) lamp this results in an efficiency (in 1945) of 14 lumens per watt, where 7.5 per cent. of the input energy is converted into radiation to which the human retina responds. Recently, however, the generation of radiation has suddenly expanded into different and often spectacular methods. Reproducing Nature's cold light has long been emulated, for some 90 per cent. of the radiant emission from the firefly is luminous, closely approaching 500 lumens per watt. The high efficiency electric discharge or ionization lamps—the illuminants transferring electrical energy directly into visible radiation (or to ultraviolet) without incandescence—all these are the subjects of an intriguing story of which only the first chapter has been written.

Electrons that have been momentarily *freed* rather than merely *excited* promise us light sources for tomorrow's use that inherently possess a vastly greater variety of color quality, or a tremendously wider range of wave-lengths, and which, in the region of eye sensitivity, have already achieved luminous output efficiencies easily three times that of the Mazda tungsten filament lamp. The limit of the latter is about 54 lumens per watt at the melting temperature of tungsten—3655 degrees absolute—but practically only 35 if the lamp life is to be several hours. But with fluorescent lamps, for example, we have, after less than a decade of development, attained to more than 60 lumens per watt, and to a higher figure with mercury vapor lamps in quartz bulbs.

* From an address to the Franklin Institute, October, 1944.

New light sources thus are being born under the urge to secure greater volume and more varieties of light, and illustrative equations aiding an appraisal of the situation would be:

1. Firefly, bioluminescence, $\text{Ratio of } \frac{\text{luminous energy}}{\text{radiant energy}} = 90\%$
2. Electric power generation, $\text{Ratio of } \frac{\text{electrical energy}}{\text{coal energy}} = 30\%$
3. Fluorescent lamp radiation, $\text{Ratio of } \frac{\text{internal ultraviolet}}{\text{input (socket) energy}} = 55\%$
4. Fluorescent luminosity, $\text{Ratio of } \frac{\text{luminous energy}}{\text{elec. input energy}} = 19\%$
5. Luminous (incandescent) radiation, $\text{Ratio of } \frac{\text{luminous energy}}{\text{elec. input energy}} = 8.0\%$
6. Total (incandescent) efficiency, $\text{Ratio of } \frac{\text{luminous energy}}{\text{coal energy}} = 1.0\%$

The first prenatal study in the birth of illuminants should consider the usage intended since roughly half of the new "illuminants" or radiation emitters are intended for other than seeing tasks. In short, the "lighting" is not intended for use directly by the human eye but rather for accomplishing other tasks. The implications of this important fact justify our present review of illuminants and our consideration of the directions in which developments are trending.

Until recent years, the radiant output from a light source was of value only in substantially the one octave of wave-lengths comprised between deep red light of 8,000 A and deep violet of 4,000 A.* The peak of radiant (but non-luminous) emission of a hot black body glowing dull red is at about 40,000 Angstroms and of an incandescent tungsten filament at normal (3,000° K.) temperature at about 9,600 A. Very little of the energy radiated by a tungsten filament, therefore, lies within this luminous octave. The human eye, it will be recalled, is limited and non-uniform in its response to this octave of wave-lengths and has its peak sensitivity, under full strong illumination (photopic vision), at about 5,550 A. Under very weak illumination on the order of .001 footcandles or less, this sensitivity peak is at 5,070 A (scotopic vision of the dark adapted eye), and color is not a factor—merely brightness. Thus if detailed vision were our sole interest, the optimum color of work light, especially for signals or beacons, might be yellowish-green, but habits of appearances are against this quality if we consider that the objective is the illumination of targets or tasks.

* A, abbreviation of Angstrom, is the lineal dimension of a ten millionth of a millimeter.

Many new illuminants being developed will expand into the two octaves adjacent to the eye-exciting one and thus may be intended to produce useful radiation of wave-lengths as long as 16,000 A and as short as 2,000 A. In essence, we have trebled the range of usefulness of radiation produced by "light sources" and while the devices producing infra-red light (or a quality of emission longer than 8,000 A wave-length), or producing ultraviolet light (shorter in wave-length than 4,000 A) should not technically be called "illuminants," yet these radiators are so closely allied with devices that we have heretofore called lamps that we can simplify our classifications if we term them illuminants.

Arching across tomorrow's scientific sky is a great radiant energy rainbow or electromagnetic spectrum which roughly includes some 88 octaves of wave-lengths stretching all the way from those that are several miles long (radio broadcasting) to those which are one billionth of an Angstrom in length (the secondary cosmic rays). The three octaves of wave-lengths (16,000 A to 2,000 A) within which most of tomorrow's light sources are being born thus might be thought of as lying roughly in the center of the great spectrum and comprising a very small fraction of its range.

INFRA-RED RADIATORS OR LONG-WAVE EMITTERS.

Stimulated by military needs, we might note that several infra-red "illuminants" have recently been born and are doing remarkable service in connection with the production of war materials. Outstanding among these are the groups of drying lamps or heat lamps with tungsten filaments operated at some 2,500° K. The peak energy output of this family of radiators is aimed at about 10,000 to 12,000 A because at this wave-length we find excellent penetration through films of transparent liquids and into human tissue. The results in industry include quick deep drying of lacquers and varnishes and evaporation of water, or the annealing and softening of solids. In human or animal therapy there is a species of heat erythema from an enlargement of the capillary blood vessels and a resultant increased circulation or relief from congestion which is frequently beneficial to health.

These radiant heat lamps are expanding into types whereof the absorption coloring of the glass bulb or of an accessory filter removes much of the visible light, hence giving us new tools both for human therapy or—of greater commercial importance—localized heat radiators for human comfort in tomorrow's residences. Experimental units are already developing for placement above a bed as a substitute for one or more blankets; for mounting flush with the garage floor to facilitate cold weather starting of automobiles; to be used as supplemental to the ordinary electric range or cooker in the preparation of foods; for dehydrating vegetables; to humidify by generating steam, or generating

pleasant odors; clustered in the fireplace as a substitute for burning logs, and in plant growth or animal husbandry.

NEW SOURCES IN THE REGION OF THE VISIBLE SPECTRUM.

Variations of hot wire illuminants may be characterized by those recent lamps developed primarily for resistance to gun fire and shock

FIG. 1. Perfections of incandescent lamps are attained through the medium of projecting an enlarged filament image upon a screen.

as on battleships or in tanks whereof the filament wire is peculiarly tough and the glass bulb remarkably shatter-proof. Another example is the modern diving lamp, the diver's gargantuan spotlight for underwater operations only. This lamp epitomizes much of the manu-

facturer's art. It makes use of a heat resisting, crystal clear, non-collapsing glass bulb,* a remarkably concentrated double coiled filament of extreme brilliance, the concentration of 1,000 watts in a bulb only some 3 inches in diameter, and the use of a thimbleful of tungsten crystal sand within the bulb which scrubs away the deposited blackening when the bulb is agitated.

The aiming of big guns, independent of the glare conditions of a target, has recently been made possible through the so-called gunsight lamps with concentrated filaments focussed in side-silvered bulbs. Similar lamps emitting beam candlepowers of 1,200 but consuming but

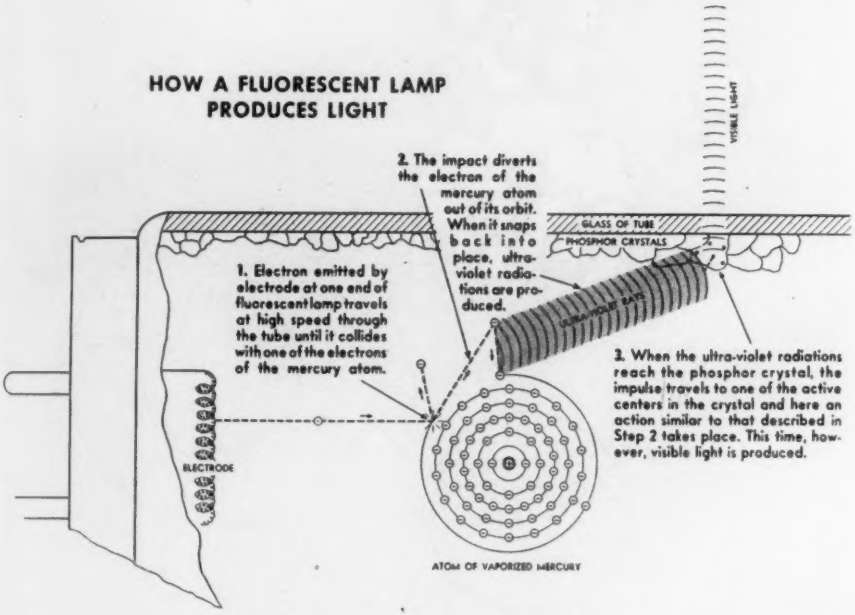
HOW A FLUORESCENT LAMP PRODUCES LIGHT
VISIBLE LIGHT
2. The impact diverts the electron of the mercury atom out of its orbit. When it snaps back into place, ultra-violet radiations are produced.
GLASS OF TUBE
PHOSPHOR CRYSTALS
1. Electron emitted by electrode at one end of fluorescent lamp travels at high speed through the tube until it collides with one of the electrons of the mercury atom.
ULTRA-VIOLET RAYS
3. When the ultra-violet radiations reach the phosphor crystal, the impulse travels to one of the active centers in the crystal and here an action similar to that described in Step 2 takes place. This time, however, visible light is produced.
ELECTRODE
ATOM OF VAPORIZED MERCURY

Fig. 2. The generation of light in a fluorescent lamp is a dual process,—first the ultraviolet radiation from warped or dislocated vapor atoms, and second the transformation from short-wave to longer-wave radiations by the phosphor crystals.

6 watts have greatly aided sea-rescue operations. Many adaptations of known methods have been made to wartime devices such as the use of krypton gas within small bulbs to blanket the heat and increase luminous efficiency (flashlight types) or helium and hydrogen filled bulbs to give quick cooling of filaments and expedite signal flashing.* Then, of course, there is the lusty family of young illuminants, the fluorescent lamps.

* Withstanding 300 pounds pressure per sq. in.

* Heat losses decrease as atomic weights of gases used to surround filaments increase. Helium is 3.99; nitrogen 14.01; argon 39.88; and krypton 82.92.

NEW FLUORESCENT ILLUMINANTS.

No tale of the birth of a light source could be more illustrative of prolonged determined research and the slow spiraling advancement of knowledge than of this epochal development. Fluorescent materials were known centuries ago—barium sulfide in 1602 and calcium sulfide in 1768. The fact that an electric discharge in certain gases or metallic

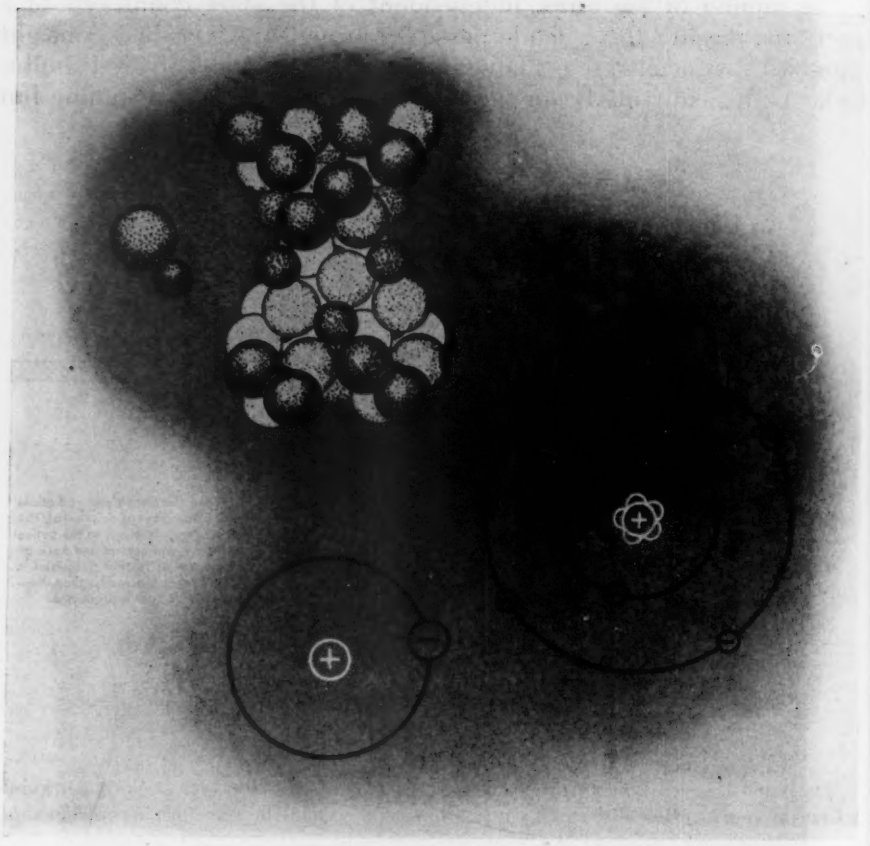

FIG. 3. The atom may be likened to a minute solar system wherein the electrons are planets. When ionized, some of the outermost electrons are displaced and radiant energy is emitted when they return to normalcy.

vapors such as argon or mercury would develop ultraviolet radiation to excite these fluorescent compounds has also been known longer than the working of tungsten for filaments. Strange, then, that the fluorescent lamp was so long a-borning. Fortunate indeed was the development of special crystal forms and regulated impurities of inorganic tungstates and silicates to the point where these could convert nearly one third of the incident ultraviolet energy to visible radiation. These so-called

phosphors respond to the excitation of a major short wave line of ionized mercury vapor, namely 2,537 A. More specifically, the absorption curve of natural willemite and of its synthetic brother, zinc silicate phosphor (Zn_2SiO_4), peaks at approximately 2,500 to 2,700 A. In close conformity, the excitation curve of this phosphor peaks at some 2,600 A. Happy also was the perfection of electron emission coatings on cathodes which provides the machine-gun stream of "bullets" or freed electrons that ionize the mercury vapor, plus the means of bal-

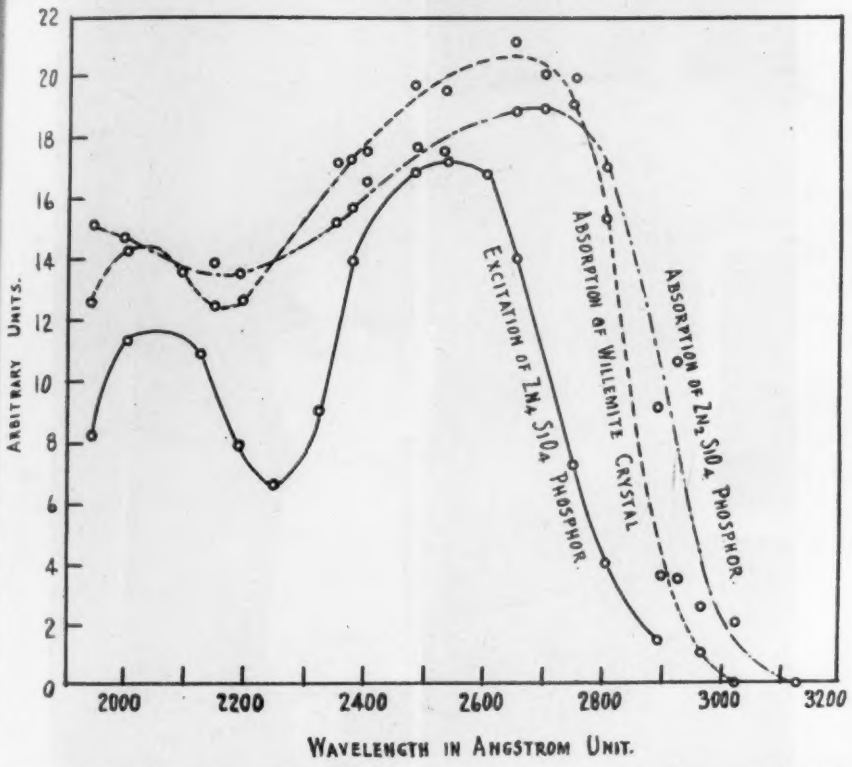

FIG. 4. In developing luminescent phosphors, the studies involve crystal or molecular forms and resonance. Shown here are the responses of a typical fluorescent lamp material.

lasting an unstable electric arc. With all the technique of proper proportion of tube diameter to arc length, all of this research providentially culminated in convenient light sources desperately needed to illuminate the great war industries. This electronic (fluorescent) lamp came of age barely in time to greatly help the avoidance of an industrial electric power shortage.

The fluorescent lamp further emphasizes the fact that, like humans, the fledgling lamp is continuously growing and developing. According to our best calculations, the present day phosphors are considerably less

efficient in converting short-wave to long-wave radiations than they should be. Of the 40 watts of power input into a modern fluorescent Mazda tubular lamp, some 22 watts are translated into ultraviolet radiation (kept within the glass tube) of wave-length chiefly 2,537 A. When impinging upon the phosphor coating, some 6.5 watts are con-

FIG. 5. New shapes of fluorescent lamps are influenced by efficiencies and by architectural usage—in this illustration a lamp eight feet long.

verted to visible light which, when added to the direct visible fraction of blue light coming from the excited mercury vapor and leaking through the interstices of the coating, finally provide the equivalent of about 7.5 watts, or 18.5 per cent. of the input energy converted to the usable light. We should do better.

Three points of attack are being pressed home as the fluorescent lamp becomes of age:

A—Studies of crystal structure and grain size of phosphors to increase their permanent ability to transform short-wave ultraviolet into visible light. Relationships between metallic "impurities" or activators, and the bonding of atoms seem a vital part of fluorescent efficiencies. Thickness and uniformity of coating is another factor.

B—The generation of more short-wave ultraviolet per watt within the tube, one of these objectives being the lowering of the per cent. of the voltage drop at the electrodes. Independence of the influence of outside temperatures, plus optimum ratios of watts per foot of length and per inch of cross-section, and exacting amounts of mercury or of mercury vapor represent some of the studies.

C—Commercially desirable forms or shapes of tubing, notably the so-called Slimline lamps up to 8 feet in length and 1 inch diameter and the Circline lamps in the form of a torus with diameters of $8\frac{1}{2}$, $12\frac{1}{4}$ and 16 inches.

HOW HIGH INTENSITY MERCURY LAMPS ARE DEVELOPING.

There is a reasonable urge to perfect the high intensity mercury vapor lamp because (1) it is currently capable of producing better than 60 lumens per watt and (2) it has shown excellent life performance on the order of 5,000 to 10,000 hours of burning. But there are still some unknowns to be solved—new sizes to be added, and high pressures of the mercury vapor to be reached before this lamp approaches a vigorous youth. It was born about 1925. Today a major objective is to perfect the use of quartz as an inner bulb in order that through higher temperatures and pressures we can achieve whiter light and more efficient output. A concurrent question is by what means can we reduce the voltage needed to start the arc in cold weather when mercury vapor condenses, and when we have not yet discovered an optimum electron emitting material that can be used in quartz without detriment thereto.

Allied to the mercury vapor lamp is a new illuminant that in a semi-practical form is being demonstrated to the Franklin Institute for the first time. This is the tellurium vapor lamp with the following remarkable characteristics:

1—It is a lamp operating essentially on direct current of medium to high voltage.

2—It is a source which, contrary to black body radiators, has a bluish color at low temperature when starting, and develops a golden yellow color at operating high temperatures.

3—Contrary to the usual experience with broken spectra of excited metallic vapors, it emits a continuous spectrum from its vaporized tellurium metal.

NEW ULTRAVIOLET LAMPS.

Supplementing the fairly recent progress stories of Sterilamps or bactericidal lamps known to this Institute have now been added the

FIG. 6. Wireless power akin to Radar is capable of ionizing metallic vapors and thereby producing fluorescent radiation.

developments of glass (vicor) capable of unusually high and unusually sustained transmission down to wave-lengths as short as 2,000 A. New shapes of tubing include one that in the form of a shoetree may sterilize

the linings of shoes as against the fungi of athlete's foot, or in a narrow U shape for insertion in bottles and jars. Another is a short-wave ultraviolet emitter known as the "thin window" lamp; namely, a bulb with a very thin glass area achieved by an inblown bubble of glass at the end of a regular bulb. This lamp is finding an expanded application in control of wound infections and skin diseases.

Added to the list of ultraviolet sources must be included an expanded development of black light illuminants—namely those emitting essentially 3,600 A radiation and which have graduated from the job of stage entertainment and of fluorescent toys to the greater job of crime detection, analyses of glass, of chemicals, of dust collected in clothing or foods, disclosure of ringworm and other diseases, as a sure test for death, and many medical diagnoses and including the blacklight floodlights on the airplane carriers or flat tops. Here the signal officer with his fluorescent flags is, under the black light, easily visible to the air pilot, meanwhile being invisible to the enemy airmen at greater ranges.

TOMORROW?

Wireless power or short-wave energy akin to radar and even up to 20 meters wave-length is being studied as a means of exciting peculiar illuminants to a reasonably good light emission without the need of connecting wires or metallic circuits. The scientist dreams of luminescent pigments in paints and dyes—of tinted surfaces that reflect literally more visible light than falls upon them. The engineer speculates on the unsolved problems of how to get illumination without glare, or how to enrich our night-time living through liberal varieties of colors in our illumination. Whatever the method, new illuminants are being born and brought up with surprising rapidity, far exceeding any other richness of development during man's entire history. One suspects that we are witnessing only the prelude of the radiant drama.

Moisture and Keeping Quality in Dehydrated Vegetables.—Keeping quality of dehydrated vegetables increases markedly with a decrease in the moisture content, according to tests made at the Western Regional Research Laboratory of the U. S. Department of Agriculture.

Dehydrated vegetables lowered in moisture content beyond present commercial attainments have been found to retain their quality proportionally better, without refrigeration as a protective measure, according to L. B. Howard of the laboratory staff. Although current dehydrated vegetables are serving their purpose, they are still subject to deterioration during storage, especially if they must meet the hazard of high temperatures for prolonged periods. Consequently there is need for an efficient and feasible method of lowering the moisture content below present purchase specifications.

Still lower moisture contents are attainable only through a limited number of measures, Dr. Howard points out. These would require additional costs. Of the methods now available, the use of the "in package" desiccant, or drying agent, such as calcined lime held in moisture-permeable packages, seems to offer the most promise.

A package of desiccant placed in a can with dehydrated vegetables and held there under seal will continue to remove moisture from the dried vegetables during storage and transportation. In the tests at the Western Laboratory, sufficient time has not yet elapsed to indicate appreciable deterioration in experimental packages held at room temperature and containing desiccating material. So-called accelerated tests—in which packages containing desiccant were held at temperatures much higher than room temperature—have indicated that several years may elapse before appreciable deterioration will occur in those held at common storage temperatures. Details of mechanical handling of packages containing desiccant material are being worked out.

R. H. O.

NOTES FROM THE NATIONAL BUREAU OF STANDARDS.*

DIAMOND CUTTING ACCELERATED BY ELECTRIC ARC.

The method universally employed for cutting plane surfaces or facets on diamonds has been to place the diamond in contact with a flat cast iron lap, charged with diamond powder and rotated at about 2,000 revolutions per minute. As a general rule, cutting is most rapid when the plane of the facet is parallel to one or two of the crystal axes of the diamond and the direction of motion of the lap surface is parallel to one of the axes. For other orientations, the cutting rate decreases until, on an octahedron face which is approximately at 35° to all three axes, cutting is next to impossible. C. G. Peters, chief of the Bureau's Interferometry Section, has found that by producing a high voltage electric arc at the contact between the diamond and lap, the cutting rate is very materially increased for all orientations of the facet. The dop that holds the diamond is connected to one, and the rotating lap to the other secondary terminal of a 5,000 volt power transformer. When 110 volts A.C. is applied to the primary of the transformer and the primary current is adjusted to about 0.5 ampere, a small blue arc is formed at the contact of the diamond and lap.

The arc about doubles the cutting rate for the most favorable directions and increases the rate about four times for the least favorable directions on the cube and dodecahedron faces. The cutting rate on the octahedron faces is about the same as that for the least favorable directions on the cube or dodecahedron faces. By applying the arc to a diamond saw, the sawing rate is greatly increased and diamonds can be sawed regardless of the orientation of the cut relative to the crystal axes.

A complete description of the method will appear in a forthcoming number of the *Journal of Research*.

EFFECT OF POLISHING THE BASE METAL ON PROTECTIVE VALUE OF ELECTROPLATED COATINGS.

It is a common belief that if metals are plated upon a rough metal surface, the deposits are likely to be more porous than on smooth surfaces, and hence to furnish less protection against the corrosion of the base metal. An investigation was conducted by Gerald A. Lux, research associate of the American Electroplaters Society, and William Blum of the Bureau's electrochemistry section to determine what effects

* Communicated by the Director.

are produced upon the protective value of coatings by differences in the degree or method of polishing the base metal.

The first phase of the investigation, which is reported in the April *Journal of Research* (RP 1645), was confined to the plating of copper, nickel and chromium on strips of cold-rolled steel that had been polished, principally with glued abrasive wheels having abrasives of known grain size. The specimens were then plated with definite thicknesses of copper, nickel, and chromium and exposed to the atmosphere in New York, N. Y.; Sandy Hook, N. J.; and Washington, D. C. They were inspected at intervals and rated according to the extent of rust.

The steel was polished with abrasive grains from 90 to 320, and some specimens had a "superfinish," applied by polishing with fine abrasive and oil. The surface finish on the various polished steel specimens was measured with a Profilometer and expressed as the root mean square of the depth of scratches, in microinches. Valuable data were obtained regarding the effects of polishing upon the surface finish and upon the amounts of metal removed.

The exposure tests showed that the wide differences in the surface finish produced practically no effects upon the protective value of the coatings. Results in accelerated tests, such as the salt spray, were less reproducible and consistent than in the atmospheric tests. The failure to observe effects of polishing may be accounted for by the use in this study of a high grade of cold-rolled steel that had less inclusions than may be present in hot-rolled steel. It is planned to extend the study to include other grades of steel, and also brass and zinc-base die-castings.

CATHODIC PROTECTION OF PIPE LINES.

In order to study the effectiveness of zinc anodes for the cathodic protection of iron and steel under various soil conditions, zinc-iron couples were buried at eight test sites under the Bureau's auspices in 1941. Measurements of the current and potential were made on the couples at the time of installation and at subsequent inspection periods. In general, the currents have been maintained during the period covered by the tests. At two sites, the current required to protect the iron cathodically was measured. Couples were removed from two sites and cleaned, after which the loss in weight of each element was determined. Coupling to zinc reduced somewhat the corrosion of the iron for 1.5 years in a soil containing sodium carbonate, but the degree of protection cannot be considered adequate. At the other site the iron was almost completely protected galvanically during the exposure period of 3.14 years.

The complete report on this work, by Melvin Romanoff, will be published in the *Proceedings* of the National Association of Corrosion Engineers.

FIRE EXPOSURE TESTS OF BUILDING CONSTRUCTIONS AS RELATED TO ACTUAL FIRES.

In the standard fire resistance test of building constructions, the prescribed furnace temperatures are generally at higher levels than for fires that occur in actual buildings. However, it has been indicated that substantially the same temperature rise is obtained on the unexposed side of a construction if the product of time by temperature, that is, the area under the fire exposure curve, is the same. To obtain further information on this point, a refractory slab was subjected to carefully controlled exposures to uniform temperatures in the range 700° to 1,300° F., and the rise in temperature on the unexposed side noted. It was found that within this range a rise of 250° F. on the unexposed side was obtained for very nearly the same time-temperature area, the variations being within experimental error.

It was also indicated by the results (as also from theoretical considerations) that, assuming no change in the thermal properties of the construction, the ratio of temperature rise on the unexposed surface to hot-side temperature is a function of time only and does not depend on temperature. Hence it was possible by mathematical methods to apply the results obtained at constant hot-side temperatures to the solution for conditions where these vary with time as they do in furnace tests and in exposures of constructions to fires in buildings.

SOUNDNESS TEST FOR HYDRATED LIMES.

A few years ago, as a result of an investigation at the Bureau of a particular kind of plaster failure which is characterized by the formation of bulges in the finish surface, manufacturers of lime started to convert their hydrating plants to produce a finishing and masonry lime in which the magnesia was more nearly completely hydrated. The war, with its demand for steel and other critical materials, has delayed the conversion. Therefore, there is still a need for a performance test to differentiate hydrated limes that can cause a marked expansion from those giving little or no expansion. An investigation has been undertaken to develop an accelerated autoclave test for this purpose and the test procedure has been formulated.

Fifty-seven different samples of hydrated lime, representing a fair cross-section of the types produced, have been mixed with one brand of portland cement, cast into bars, and the expansions of the bars determined. Very good checks have been obtained by two different operators using two different autoclaves. Other brands of cement are now being used. Although, in general, the bulk of the limes, when tested with different cements, give expansions in sufficiently close agreement after correcting for the expansion of the cement, there are a few limes

that, with different cements, give expansions having a greater spread than is desired. It is hoped that the cause of this can be found and that this spread can be eliminated or reduced.

MELTING POINT OF ALPHA-ALUMINA.

The value usually quoted in the literature for the melting point of alumina (Al_2O_3) is 2,050° C., which was published by C. W. Kanolt in 1914. Several investigators have since published values ranging from 2,001° C. to 2,045° C.

Alumina is not only an important constituent in many industrial products but has become essential in super-duty spark plug insulators, refractories for special high temperature applications, and also for insulators in the field of electronics. Also, a large number of systems which have been, or are being, investigated for the determination of phase relations include alumina as one of the end members. Consequently, it was considered important to attempt again the establishment of the true melting point of alumina of high purity and under carefully controlled conditions.

Three samples were available, all of which contained over 99.9 per cent. of Al_2O_3. The specimens were heated in an oxidizing atmosphere, and under atmospheric pressure, using a furnace in which an electric current, passed through resistors made of thoria and ceria, produced the necessary temperature. The heating rate ranged from 1° C. to 5° C. per minute and observations were made with an optical pyrometer.

In seven tests, the minimum temperature at which melting was seen to begin was 1,990° C. and the maximum was 2,010° C. Although spectrographic analyses of four of the specimens, made after the melting tests, showed that they had "picked up" measurable impurity from the furnace atmosphere, especially silica and magnesia, the indications are that these melted materials contained about 99 per cent. of Al_2O_3.

As a result of these tests, which were made by R. F. Geller and P. J. Yavorsky and are reported in the April number of the *Journal of Research* (RP 1649), it is believed that alumina melts within the range 2,000° to 2,030° C.

FREEZING TEMPERATURE OF BENZOIC ACID AS A FIXED POINT IN THERMOMETRY.

Exact measurements of temperature in the range of nearly − 200° to nearly + 700° C. are based on the change with temperature of the electrical resistance of a coil of fine platinum wire; the device used for this purpose is known as a resistance thermometer. To standardize the instrument, measurements of its resistance must be made at three

fixed temperatures if it is to be used in the range above zero and at four temperatures if the lower range is also to be covered. The four fixed temperatures commonly used for the standardization are the normal boiling point of oxygen, − 182.97° C., the melting point of ice, 0° C., the normal boiling point of water, 100° C., and the normal boiling point of sulfur, 444.60° C.

For many purposes, the melting point of ice and the normal boiling point of water are the more important of these fixed points. Of these two, the melting point of ice can be quite easily reproduced with high accuracy, but accurate observations of the boiling point of water are much more difficult to make. This is because boiling points fluctuate markedly with changes in barometric pressure. Even with the painstaking barometric measurements practiced in standardizing laboratories, the "steam-point" cannot be observed as accurately as the "ice-point." In distinction from boiling temperatures, melting or freezing temperatures are only slightly affected by variations in barometric pressure. Therefore, substituting a suitable freezing temperature for the "steam-point" would simplify the standardization of resistance thermometers and, if such a temperature were sufficiently reproducible, might improve the accuracy of standardization.

In *Technical News Bulletin* No. 315 (July 1943) a preliminary report was presented on the use of the freezing point of benzoic acid as a thermometric fixed point. Results of subsequent work by Frank W. Schwab and Edward Wichers, including extended observations on the freezing point, and suggestions for the use of this point in the calibration of thermometers instead of the "steam-point," are presented as RP 1647 in the April *Journal of Research*. The purified acid is contained in a sealed glass cell provided with a thermometer well in which the instrument to be standardized can be inserted. When the acid is melted and then allowed to start freezing in the cell in a suitable manner, a very constant temperature is maintained while the acid freezes. The freezing temperature is 122.36° C. It may differ by a few thousandths of a degree between cells of a group, depending upon the purity of the acid and the conditions existing within the cell, but is fixed to within one-thousandth of a degree in a particular cell. Plans have been made to issue this type of cell with a certified temperature for use in calibrating thermometers in scientific and industrial laboratories.

The paper includes observations on the freezing temperature of benzoic acid under various conditions, on the change in volume which occurs when the acid freezes, on the solubility of oxygen and nitrogen in the liquid acid, and on the stability of the acid during prolonged heating at temperatures above its freezing point. There is an appendix, by Frank W. Schwab and E. R. Smith, on a method of calculating temperatures from observations of the resistance of platinum thermometers.

VINYL PLASTIC SHOE SOLES.

The public has heard and seen a great deal of plastic soles during World War II. The critical supply situation with respect to sole leather, attributable to increased demands by the military services and dependence to a large extent on foreign sources for hides and tanning materials, has made it necessary to seek replacements wherever possible. One type of material which has been found to have suitable properties for this application is vinyl resin. Unfortunately this synthetic material is also in very critical supply. However, scrap available from the manufacture of insulation for wire and cable has been employed for the production of shoe soles.

Twenty-two vinyl plastic materials for shoe soles were evaluated by G. M. Kline, P. A. Sigler, and P. Plaia at the Bureau in an investigation undertaken for the War Production Board. The materials were obtained from various manufacturers during the period December 1943 to February 1944. The properties measured were thickness, density, tensile strength, elongation at break, stitch tear, change in thickness on immersion in water, loss in weight on heating, flex-fatigue life at 32° F., resistance to cracking on bending at 0°, − 20°, and − 40° F., and abrasion resistance.

The vinyl plastics were inferior to leather in tensile strength and stitch tear, but were superior to leather in change in thickness on immersion in water and abrasion resistance. On the basis of these tests it is concluded that the more satisfactory grades of vinyl plastic shoe sole materials would meet the following requirements: Tensile strength (Minimum), 1,500 lb./in.^{2}; Stitch tear, dry and wet (Minimum), 30 lb. irrespective of thickness of material; Flex fatigue life at 32° F. (Minimum), 1,000,000 cycles; Bend test at 0° F., aged and unaged (Minimum), 10 bends over a 1/2-in. diameter mandrel; Abrasion resistance, leather machine (Minimum), 6,000 rev./mm.; Abrasion resistance, Taber machine (Maximum), 75 mg./1,000 rev.

The materials meeting the above requirements did not contain fibrous fillers. This is particularly worth noting in view of the many cracking failures that have been experienced with soles made of large amounts of fibrous materials bonded with small percentages of resin. These so-called "plastic" soles are akin to the rabbit meat made with one rabbit and one horse.

If the sole is to be attached to the shoe by stitching only, the stitch tear requirement of 30 lb. may not be adequate. Higher stitch tear strength can be obtained by the use of a fabric backing material.

THE FRANKLIN INSTITUTE.

MEDAL DAY MEETING.

WEDNESDAY, APRIL 18, 1945.

The annual Medal Day exercises at The Franklin Institute began at 5:30 on the afternoon of Wednesday, April 18th. Members and friends of the Institute who were unable to attend the dinner were present for the presentation of awards which began at 8:15 p.m.

After a reception given by the Hostess Committee, the Medalists, Officers and guests were seated at dinner at 6:30. At 8:15 the program of the evening was started, with Mr. Charles S. Redding, President of the Institute, presiding.

He asked the audience to stand in silent tribute in honor of our late President, Mr. Roosevelt. Immediately after, the National Anthem was played by Lieutenant Guy Marriner, AUS, a member of the staff of The Franklin Institute, on leave in service.

The President then announced that as this was the regular monthly meeting of The Franklin Institute it was in order to call for approval of the minutes of the February meeting which were published in full in the March JOURNAL. As there were no corrections the minutes were declared approved as printed.

The program and list of awards of medals follows:

PROGRAMME.

Reception to Medalists ..... THE HOSTESS COMMITTEE

Dinner ..... MEDALISTS, OFFICIALS AND GUESTS

NATIONAL ANTHEM ..... GUY V. R. MARRINER
1st Lieutenant, A.U.S.

Toast to Benjamin Franklin ..... RICHARD T. NALLE
Chairman, Board of Managers' Committee, National Franklin Committee

Greetings ..... CHARLES S. REDDING
President, The Franklin Institute

Introduction of Medalists of Other Years ..... THE PRESIDENT

"Another Year" ..... HENRY BUTLER ALLEN
Secretary and Director, The Franklin Institute

Presentation of Awards ..... THE PRESIDENT

"On the Natural History of Globular Star Clusters" ..... HARLOW SHAPLEY

"America" ..... GUY V. R. MARRINER
1st Lieutenant, A.U.S.

MEDALISTS.

Presentation of Certificates of Merit

WALTER J. COPPOCK
Associate Professor of Industrial Engineering
Pennsylvania State College
State College, Pennsylvania

GREER ELLIS, Manager
Stresscoat Division
Magnaflux Corporation
Chicago, Illinois

Presentation of Levy Medal

RUPEN EKSERGIAN
Budd Manufacturing Company
Philadelphia, Pennsylvania

Presentation of Potts Medal

EDWIN A. LINK, President
Link Aviation Devices, Incorporated
Binghamton, New York

Presentation of Longstreth Medal

SANFORD L. CLUETT, Vice President
Cluett, Peabody and Company, Incorporated
Troy, New York

Presentation of Cresson Medals

STANFORD C. HOOPER, Rear Admiral
United States Navy (Retired)
Washington, D. C.

LEWIS F. MOODY
Professor of Hydraulic Engineering
Princeton University
Princeton, New Jersey

Presentation of Brown Medal

GILMORE DAVID CLARKE, Dean
College of Architecture
Cornell University
Ithaca, New York

Presentation of Clamer Medal

ZAY JEFFRIES, Vice President
General Electric Company
Nela Park
Cleveland, Ohio

Presentation of Franklin Medal

HARLOW SHAPLEY, Director
Harvard College Observatory
Harvard University
Cambridge, Massachusetts

(A detailed statement of these exercises together with the paper presented by the Franklin Medalist, will appear in an early issue of the JOURNAL.)

LIBRARY NOTES.

The Committee on Library desires to add to the collections any technical works that members would wish to contribute. Contributions will be gratefully acknowledged and placed in the library. Duplicates received will be transferred to other libraries as gifts of the donor.

Photostat Service. Photostat prints of any material in the collections can be supplied on request. Orders received in the morning are filled the same day. The average cost for a print 9 × 14 inches is thirty-five cents.

The library and reading room are open on Mondays, Tuesdays, Fridays and Saturdays from nine o'clock A.M. until five o'clock P.M. Wednesdays and Thursdays from two until ten o'clock P.M.

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N. B. Effective June 1 until September 15, the Library hours will be: Mondays, Tuesdays, Wednesdays and Fridays, from nine o'clock A.M. until five o'clock P.M.; Thursdays, from two o'clock P.M. until ten o'clock P.M.; Saturdays, from nine o'clock A.M. until twelve o'clock noon.

NOTES FROM THE BIOCHEMICAL RESEARCH FOUNDATION.

An Attempt to Reproduce in vitro **Certain Aspects of Encapsulated Infection.**—ROBERT K. JENNINGS. The bacteriologist, culturing and studying the behavior of microorganisms, has always sought to reproduce the environment in which they normally operate. He knows, *a priori*, that they will thrive under these conditions, and he avoids misleading results arising from unnatural stimuli. Thus, it is common practice to maintain incubators at body temperature, employ the water-soluble extractives of animal tissues as culture media, and to adjust the pH and tonicity of the substrate to the physiological range.

Experiments now being undertaken at the Biochemical Research Foundation have as their basis a desire to study the behavior of bacteria —in particular, the staphylococcus—under the conditions of a long-standing, localized infection. For the purpose of our work, this is conceived of as a walled-in culture of organisms which do not, presumably, increase greatly in numbers but which give rise to metabolic or catabolic products which escape from the culture sac and give rise to toxic symptoms in other parts of the body. This natural culture vessel differs from the laboratory test-tube in that it is immersed in fluid which has been subjected to aeration in the lung, serves as an inexhaustible source of nutrient material if the necessary foodstuffs are capable of penetrating the sac, and also provides a means of disposal of waste material, subject to the same limitation.

It is recognized, of course, that the picture of the host as a "vessel" for the cultivation of the parasite is an oversimplification of the true state of affairs. No consideration has been given to the antagonistic elements involved—the phagocytes, antibodies and other protective mechanisms of the host. For the primary purpose of developing an understanding of the condition of the staphylococcus itself during infection, we have undertaken to reproduce first the advantages of the somatic environment. When this has been done to the best of our ability, the complicating factors may be introduced.

Rogers and Whittier (*Journal of Bacteriology*, **20**: 127, 1930) studied the behavior of *Str. lactis* in a culture from which the medium was continually withdrawn through a bacterial filter, and replaced with fresh medium. They found the population in such a vessel remained constant, indicating that dead cells did not accumulate. They postulated two possible explanations. Either the cells autolyzed at a rate which balanced the reproduction, or else they reached a state in which death and reproduction ceased. The authors favored the latter theory. They also found that fermentation continued unabated with the organisms in this state.

Le Guyon (*Comptes rendus des séances de la Société de biologie*, 106: 242, 1931) grew bacteria in a collodion sac from which the by-products of metabolism could escape by ultrafiltration. He does not, however, describe the characteristics of such cultures. A somewhat similar device, on a more elaborate scale, has been developed by Harris (*Journal of Bacteriology*, **45**: 147, 1943) with the additional feature that the collodion vessel is immersed in nutrient solution which is changed periodically. He has used this apparatus primarily to investigate factors which contribute to the initiation of growth.

The apparent analogy between such devices and focal infections led us to employ similar equipment to study the staphylococci. The feasibility of culturing the organisms under these conditions was verified by using a set-up similar to that of Le Guyon, that is, a dialysis bag containing the culture, and suspended in nutrient medium.

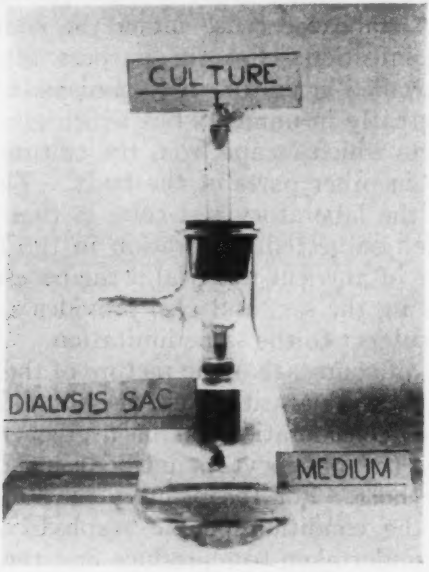

FIG. 1. A strip of cellophane tubing tied at one end is slipped over the bulb of the calcium chloride tube. The cellophane tube is filled with water, inoculated with a broth culture of staphylococcus, and suspended in broth contained in the flask. The medium can be renewed by simply transferring the culture to a new flask. (For photographic purposes the sac contains a darker fluid than that in the flask.)

A small calcium chloride tube was mounted in a rubber stopper which fitted the mouth of a 250 cc. suction flask. Cellophane "sausage casing," slipped over the bulb of the $CaCl_2$ tube and securely tied at the other end, formed the dialysis bag. The mouth of the $CaCl_2$ tube and the opening of the side arm of the flask were plugged with cotton. Both sac and flask were partially filled with water, and sterilized by autoclaving.

After sterilization the water in the flask was replaced with sterile beef heart infusion broth, which filled the flask nearly to the side arm. It was necessary to sterilize the broth separately, since so full a flask would boil over when autoclaved. The sac was inoculated with 1 cc. of a 24-hour culture of an avirulent *Staph. albus* strain in heart infusion. The level inside the sac was adjusted to that outside with sterile water. The entire apparatus was incubated at 37°. Good growth was obtained in 48 hours. The culture was permitted to develop for a week, then the sac was transferred to a fresh flask of medium. The used medium proved to be sterile. The pH was unchanged and the medium was non-toxic and non-hemolytic.

Fresh medium was used to replace the old about once a month thereafter. Following the third transfer (three months and 1 week) contamination appeared in the medium. The dialysis sac still contained a pure culture of staphylococcus, however. The organisms appeared quite normal. Very few gram-negative cells were observed and viability was unimpaired. The strain used in these tests had been isolated from normal skin. It did not produce demonstrable toxin in normal cultures, but did give rise to quite potent hemolysins. Hemolytic activity never became demonstrable in the medium, however, in spite of dense growth in the sac.

Growth was profuse. Immediately above the medium (which in later transfers was below the inner level of the culture) confluent colonies appeared on the surface of the cellophane. Below this level a comparatively clear space covered what appeared to be collected bacterial bodies which half filled the sac. The lowest part of this sediment was quite dark in color. We do not as yet know just what the pigment is. The culture itself was very viscous, as though a great deal of polysaccharide had accumulated in the sac.

From these orientational observations we conclude that the growth in the dialysis bag resembles growth on solid medium. The membrane functions as does the surface of agar, giving support to the organisms, and supplying nutritives and removing waste products by diffusion. Indeed, at the point where the sac extends above the surface of the medium growth occurs in colony form, presumably because at this point free access to oxygen is obtained, while the incoming supply of food is still great enough to meet the requirements of a large number of organisms.

It is our immediate intention to continue and extend this line of investigation. We have developed an apparatus which permits the dialysis into the culture of nutrients from a medium continually saturated with air. This, we hope, will furnish conditions over the entire surface of the membrane similar to those at the air-liquid juncture in the simpler vessel.

If we should succeed in obtaining conditions in which toxins are released from the culture into the medium we will have a "working model" of a focal infection which should yield some worthwhile information. Failure to obtain hemolysins in the medium in the preliminary experiments may be due to a number of things: the membrane may not permit the passage of this material, or the dilution of the hemolysin may be so great as to interfere with its demonstration. We intend to make use of the analytic facilities available at the Biochemical Research Foundation, such as the electrophoresis apparatus, to follow changes in the composition of the medium during its utilization, and if possible to isolate and identify toxic and antigenic products which are released into the artificial "body fluids."

BOOK REVIEWS.

FATS AND OILS, An Outline of Their Chemistry and Technology, by H. G. Kirschenbauer. 154 pages, illustrations, 15 × 24 cms. New York, Reinhold Publishing Corporation, 1944. Price $2.75.

Vegetable and animal fats and oils are becoming more and more diversified and specialized in their uses. Those engaged in the industries which utilize these materials directly or indirectly are consequently required to cope with problems of greater diversity. A condensed treatment of the fundamentals of the chemistry and technology of vegetable fats and oils would serve good purpose in this connection. The book at hand was prepared with this in mind.

There are just 121 pages of textual matter discussing occurrence and formation of fats, constituents and components, structure and classification, nature of fats and analytical methods giving the testing procedures for crude, refined and modified oils including the testing of the oil-bearing matter as well as the analysis of such by-products as hulls, press cake and ground meal. The largest space is devoted to the technology of fats covering production, purification, hydrogenation, fat splitting, sulfonation, fatty acid distillation, soaps, paint and varnish oils, uses of fats in processes and industries. There is a chapter on some important fats where they are listed alphabetically giving origin, processing and uses. Waxes are given brief mention. The appendix gives tables of historical dates, fatty acid composition of some fats, factory consumption of fats, and characteristics of fats and waxes. There is a bibliography and subject index at the end.

This is a book of facts, condensed for quick consumption as a text and especially well arranged for reference. Those interested in the subject would do well to have it at hand.

R. H. OPPERMANN.

OUTLINE OF THE AMINO ACIDS AND PROTEINS, edited by Melville Sahyun. 251 pages, tables and illustrations, 16 × 24 cms. New York, Reinhold Publishing Corporation, 1944. Price $4.00.

This work consists of a number of contributions of various authors. It is in the form of an elementary text to serve as an introduction to the subject for those who possess an adequate background for understanding. The discovery of the amino acids is the opening chapter, which reviews in quite some detail the work of researchers. This is followed by the occurrence and properties of amino acids and proteins. From both of these chapters it is seen how unique and essential a role in life processes amino acids and proteins play. A following paper on theory of protein structure presents what is in a tentative form—a layer structure which is in accord with many experiences. This is supplemented with a treatment on methods of hydrolysis, by acids, by alkalis, and by enzymes citing advantages and disadvantages which may stimulate interest in a more perfect method for hydrolyzing proteins. The synthesis and isolation of certain amino acids is discussed in a way which outlines the useful methods indicating the principles involved and pointing out advantages and disadvantages and unusual features of each of the reactions. Important applications of each method are listed and the best methods for preparing each of the naturally occurring amino acids are summarized. A paper on methods of analysis for amino acids precedes treatments on the relation of proteins to immunological reactions, theories of denaturation and detoxication, and certain thermodynamic considerations. There are two papers on metabolism, one on nitrogen equilibrium, and one on nutrition.

The interest in this subject is gaining momentum both in industry and in medicine. The revelation of new knowledge is bound to come, but will come only from a background of a good understanding of the present position. The papers of which this book consists taken as a whole

are an outline which, with the references cited, should provide an adequate basis for further study and work. The book is a contribution to progress in this field.

R. H. OPPERMANN.

THE ART OF RESUSCITATION, by Paluel J. Flagg. 453 pages, illustrations, 15 × 23 cms. New York, Reinhold Publishing Corporation, 1944. Price $5.00.

We in America are well familiar with the meaning of resuscitation, especially since the start of the present emergency. An increasing number of people are being acquainted with first aid methods of resuscitation as a public safety measure—safety in industry and in play. In short, it means the prevention by artificial respiration of asphyxial death. The British, however, have a somewhat different meaning, chiefly circulatory stimulation. With an elaboration of these definitions as a start, this book presents a framework upon which the broad experience and the many contacts of the author may readily be studied.

After giving a background on historical references to the evolution of the art of resuscitation both from the standpoint of activities by individuals as well as by organized medical groups, there is set forth the pathologic physiology of asphyxia—experimental and clinical medicine. The principles of resuscitation are next taken up. These provide a reliable index by which manual and mechanical methods of resuscitation may be appraised. The transportation of the unconscious patient is a valuable part of this section. Widespread public effects are considered, such as the air raids on London, and conditions under which unconsciousness may occur are studied with the manner of treatment. With regard to the latter, the suggestions offered for improvement are not complicated; they may be easily adopted, in whole or in part. The author then prepares a basis for the treatment of the generic problem of asphyxia and discusses the specific causes of asphyxial accidents and the indications requiring relief. Under the heading of the field of asphyxia and resuscitation there is given the work of various organizations in furthering knowledge of the subject, and the magnitude and complexities of the subject in different lines of endeavor. The chapter on asphyxia in the U. S. Army is chiefly devoted to the question of war gas poisoning, a subject of much present interest. The approach to this problem is by way of pathological physiology. A report is given as to just what happens to the respiratory and circulatory systems after exposure to various poison gases. The latter part of the book is devoted to the coordination of gas therapy—pneumatology. Appendixes cover instruction of technicians, asphyxia neonatorum, and judicial electrocution. In the back there is a bibliography and subject index.

The presentation is clear, timely and interesting. It is an up-to-date summary of knowledge of this important subject.

R. H. OPPERMANN.

PUBLICATIONS RECEIVED.

The Builders of the Bridge, The Story of John Roebling and His Son, by D. B. Steinman. 457 pages, plates, 14 × 21 cms. New York, Harcourt, Brace and Company, 1945. Price $3.50.

Photographic Enlarging, by Franklin I. Jordan. 252 pages, illustrations, 19 × 25 cms. Boston, American Photographic Publishing Company, 1945. Price $3.50.

The Future of Industrial Research. Papers and Discussion. 173 pages, 16 × 23 cms. New York, Standard Oil Development Co., 1945.

Co-operative Electrical Research. 62 pages, 19 × 25 cms. London, The British Electrical and Allied Industries Research Association, 1944.